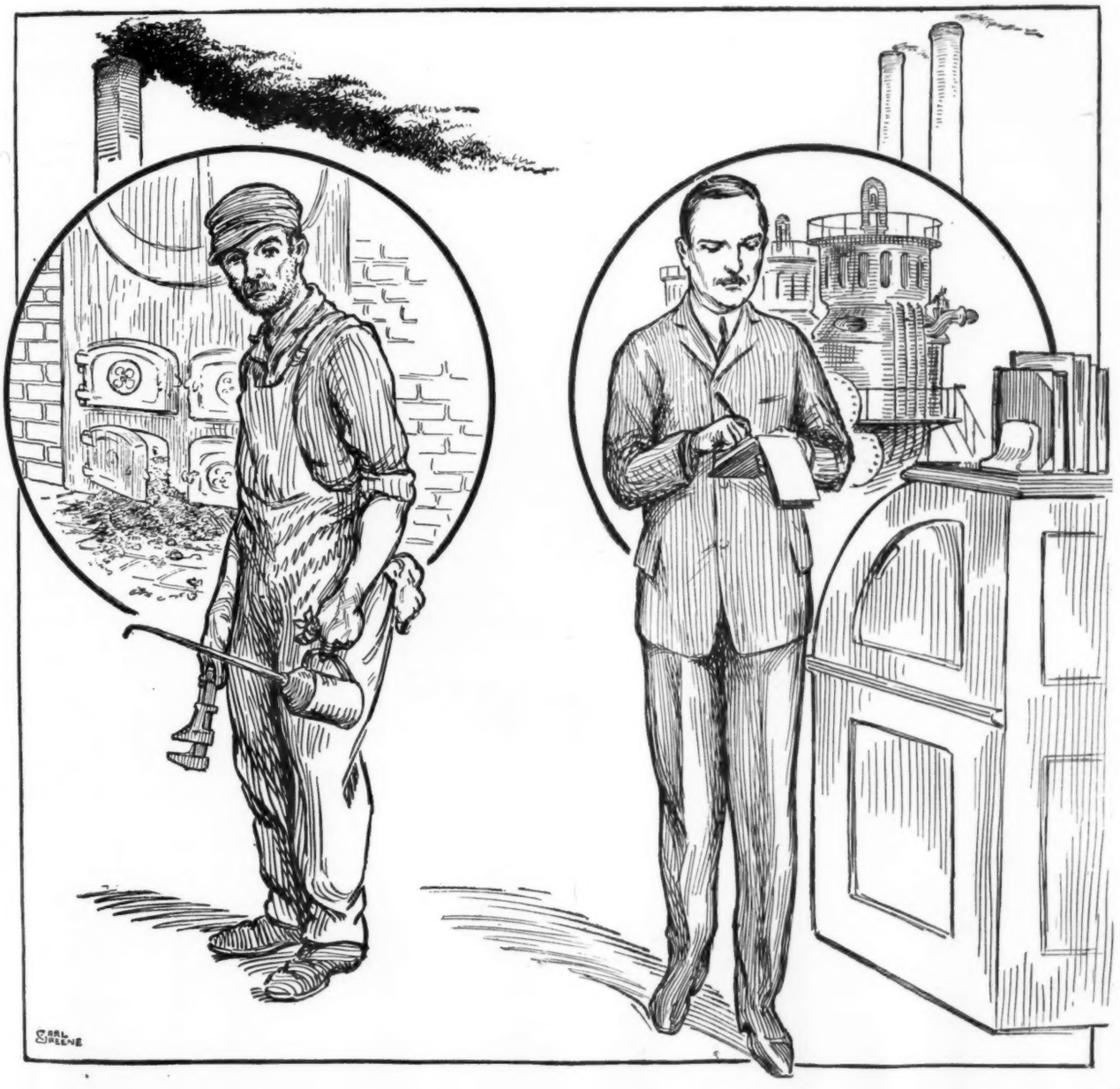
POWER

Vol. 42 NEW YORK, SEPTEMBER 21, 1915 No. 12

Twenty Years Ago and Now

All the improvement has not been in the inanimate equipment



Suggested by J. C. HAWKINS, *Hyattsville, Ind.*

Coal and Ash Handling at the Gorge Plant

By A. D. Williams

SYNOPSIS—At this plant of the Northern Ohio Traction Co., advantage is taken of its location, at the foot of a 90-ft. embankment, to provide a ready means of handling the coal by gravity from the cars to the furnaces. The ashes, after passing through a clinker crusher, are elevated by a skip and dumped into a storage bin, from which they are removed when convenient.

The Gorge Power Plant of the Northern Ohio Traction & Light Co. is located between Akron and Cuyahoga Falls, Ohio, on the east side of the gorge of the Cuyahoga River. The present installation comprises a generating capacity of 19,000 kv.-a., with a possible extension to 50,000 kv.-a. The low-water flow of the river is not sufficient to supply circulating water for the condensers, which made it necessary to construct a dam to form a storage pool, the high-water surplus from which is utilized in a 2200-kw. hydro-electric plant. There is a difference in level of 8½ ft. between the crest of the spillway and the boiler-room floor, an allowance which was justified during the exceptional spring flood of 1913, when the water rose to within less than a foot of the boiler floor. Before making the final decision as to the method of handling coal and ashes several schemes were considered. The boiler-floor level is about 90 ft. below the top of the bluff, where there is a railroad siding connecting at one end with the Pennsylvania R.R. and at the other with the tracks of the traction company. The height available below this siding afforded ample room to install a coal-storage plant of considerable capacity in one of several different ways, each permitting the coal to fall by gravity to the grates. One suggestion involved the use of a large storage bunker over the boilers, to which the crushed coal would be delivered by a belt conveyor to which it flowed from a track receiving hopper supported above the bunker, through a crusher. This arrangement was discarded owing to the heavy steelwork required to support such a storage bunker.

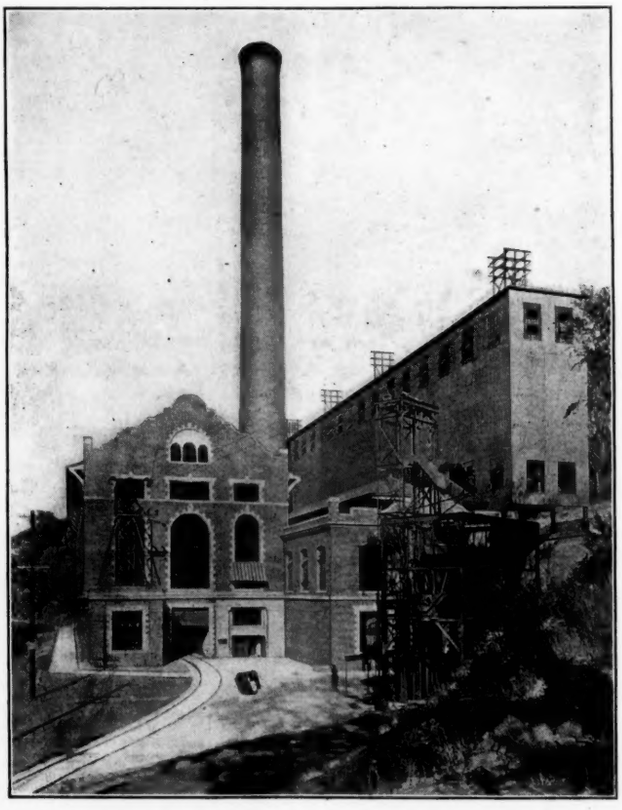


FIG. 1. SHOWING RELATIVE LOCATIONS OF POWER-HOUSE COAL- AND ASH-HANDLING EQUIPMENT

The arrangement finally adopted consists of a coal-storage structure spanning the space between the side of the boiler room and a concrete retaining wall along the embankment. This building is about 25 ft. wide and the same length as the boiler room—330 ft. At one end a steel trestle connects with the top of the bluff and carries the track, over which coal cars are pushed into the building by an electric shifting locomotive. The main storage bins are under this track, one at each end of the building, and separated by a short gap at the middle, where the chimney divides the boilers into two groups. Each bin has a storage capacity of about 1,900 tons. One of the advantages of the long track hopper is that it permits eight hopper-bottom cars to be run in and unload simultaneously. A traveling crusher runs under the receiving hoppers and can draw coal from any part of the storage bunkers and deliver it to any of the individual boiler-feeding hoppers. These hoppers are each designed to hold enough coal to operate a boiler one day. From them the coal passes through a gate into a weighing hopper, thence through a circular spout and apron spreader to the stokers. The receiving hoppers are constructed of concrete slabs supported upon a steel frame, the bottoms being sloped and forming a trough, which is closed by cast-iron gates operated from the coal crusher car. These gates extend nearly the entire length of the bin and permit the entire coal supply for the station to be drawn from any section of the receiving hopper. This feature is useful whenever any portion of the coal commences to heat.

The traveling crusher car carries a steel receiving hopper of 10 tons' capacity arranged to feed the coal to a single roll crusher from which it drops to boiler hoppers. The car is mounted on a standard-gage truck and is propelled by a 5-hp. electric motor; a 25-hp. motor drives the crusher.

Immediately below the crusher level are a number of rectangular hoppers, one for each boiler, the spouts from which pass through the boiler-room wall (see Fig. 5). Hand-operated cutoff gates here control the flow of the coal into 4,000-lb. capacity weighing hoppers, sup-

ported along the boiler-room wall opposite each boiler, and discharging into spouts to the boilers. As the coal is weighed on a track scale before the cars are dumped into the track hoppers, the amount in storage at any time may be readily determined by deducting the amount weighed out to the boilers.

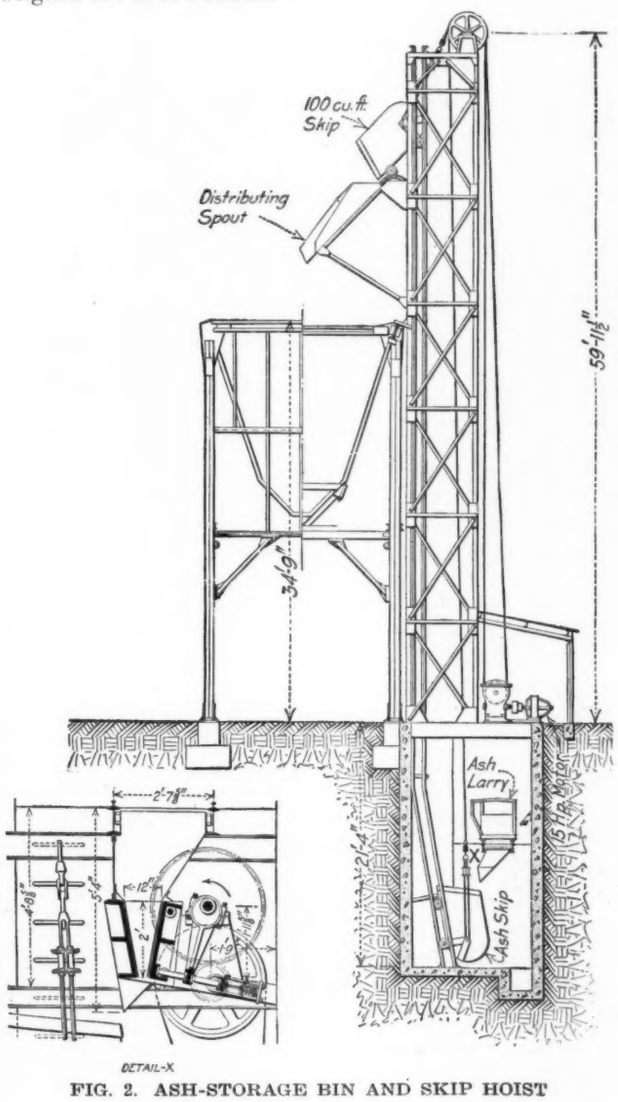


FIG. 2. ASH-STORAGE BIN AND SKIP HOIST

The slope of the spouts is a little greater than 45 deg., so that there is comparatively little tendency to choke. Each is provided with a telescopic joint a short distance above the spreader, permitting the lower part of the spout to be removed when it is desired to work at the stoker. Also, they clear the front clean-out doors, thus permitting them to be opened readily at any time.

The stokers are the seven-retort Taylor type, driven by the forced-draft fan engines through lineshafts be-

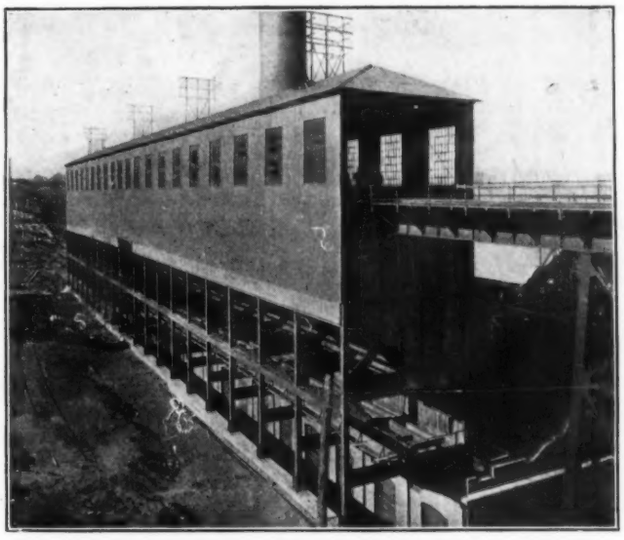


FIG. 3. COAL-RECEIVING BUNKER WITH BOILER BINS BELOW

low the boiler-room floor. Ash pockets are located under the dumping plates at the rear ends of the stokers. These are of steel plate, with cast-iron duplex cutoff gates for loading the ash larries. The gates have cast-iron renewable leaves of ribbed section.

The space below the boiler room is a shallow basement, the upper portion of one side being occupied by the blast mains from the fans and branches to each boiler, and on the other side are the ash hoppers; below these runs a 30-in. gage track for the ash larry. One of the difficulties encountered was the devising of a haulage system suited to the headroom available in the basement, the distance from the top of the rail to the bottom

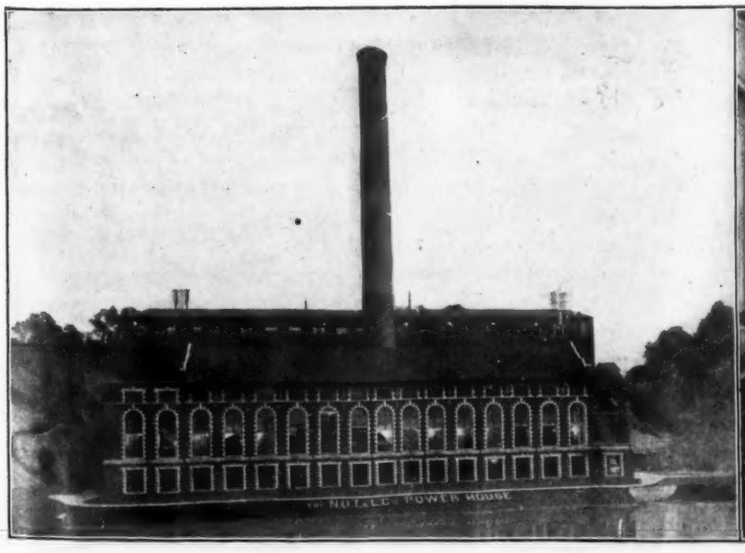


FIG. 4. SIDE VIEW OF POWER HOUSE

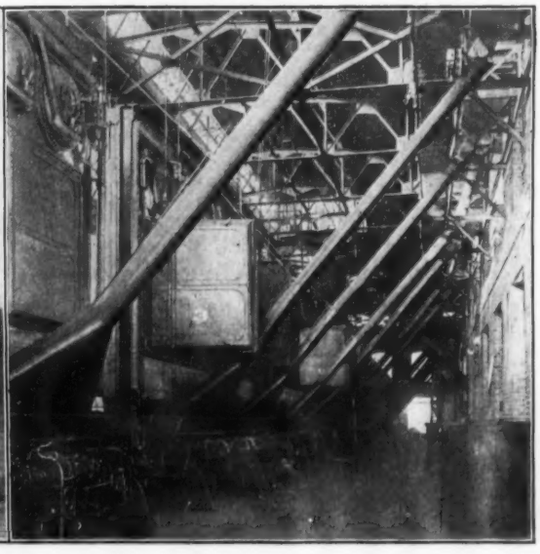


FIG. 5. FIRING AISLE, SHOWING SPOUTS

of the ash gate being only 4 ft. 4½ in. At first, the cars were moved by man-power and hauled up an incline to a temporary dumping point by a small hoisting engine. The method now in use for haulage is a small electric locomotive designed to suit the low headroom.

The ash larries are of the bottom-dump type and drop practice. The hoist in this case, however, is vertical except in the pit, where the guides are sloped to throw the skip under the ash chute. The skips dump automatically at the highest point of their travel into a breeches spout, the two legs of which serve the two ends of the storage bin. This arrangement is self-trimming,

PRINCIPAL EQUIPMENT OF GORGE POWER STATION, COAL AND ASH HANDLING

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
16	Boilers	B & W	604-hp.	Steam generation	200-lb. pressure	Babcock & Wilcox Co.
16	Superheaters	U-tube			200-lb. pressure, 75-90 deg. superheat	Babcock & Wilcox Co.
16	Stokers	Taylor	7-Retort	Boiler furnace	Driven by fan engines	American Engineering Co.
1	Chimney	Radial-brick	16x275-ft.		Natural and forced draft	Alphons Custodis Chimney Construction Co.
4	Fans			Forced draft		B. F. Sturtevant Co.
4	Engines	Variable-speed		Forced-draft fans and stokers		B. F. Sturtevant Co.
1	Coal crusher	Single-roll				Jeffrey Mfg. Co.
1	Motor	Induction	25-hp.	Driving coal crusher	440 volts	General Electric Co.
1	Motor	Induction	5-hp.	Traverse crusher car	440 volts	General Electric Co.
16	Scales			Coal to boilers	4000-lb. weighing hopper	Standard Scale and Supply Co.
1	Ash car	Bottom-dump	54-cu.ft.	Ashes from boiler to crusher	In boiler-room basement	
1	Locomotive	Electric	5-hp.	Hauls ash car		Arthur G. McKee
2	Crushers	Jaw-type	12x36-in.	Clinker crusher	In skip pit	Arthur G. McKee
1	Motor	Type-K	5-hp., 825 r.p.m.	Ash-crusher drive	550 volts, d.c.	Westinghouse Elec. & Mfg. Co.
2	Skips					Arthur G. McKee
1	Motor	Type-SK	15-hp., 500 r.p.m.	Skip Hoist	550 volts, d.c.	Westinghouse Elec. & Mfg. Co.
1	Ash bin	Baker-suspension	100-ton	Storage and car loading		Arthur G. McKee

the ashes into a crusher chute from which they pass through the clinker crusher into a 100-cu.ft. skip. There are two self-dumping skips arranged to balance each other, by means of which the ashes are elevated and dumped into a 100-ton storage bin. These skips operate in a similar manner to those used in blast-furnace alternate skip loads being dumped into each end of the bin. The bottom of the bin has four gates by which the ashes are loaded into wagons or standard-gage railroad cars and hauled away to some point where they may be used for ballasting track or wasted, the electric shifting locomotive being used for this purpose.

Will Quizz, Jr.

SYNOPSIS—The velocity of fluids in pipes forms a subject of inquiry by Will Quizz, Jr., who gets a new "slant" at the problem.

"Chief, these formulas for finding the velocity of steam or water in a pipe, the size of pipe for a given quantity delivered, etc., are not clear to me. Can the calculation and explanation be made more simple in some way?"

"Whether a problem is easy or difficult to us depends entirely on whether the subject and process are clearly understood or not. We may perform the work according to certain rules and obtain correct results, and yet know next to nothing of the subject or the reasons for the various operations. Besides, the understanding of a thing may be obtained in the most informal manner sometimes, when the regular formal statement or explanation has entirely failed to convey the idea.

"Think of a pound of steam, for instance, as a unit occupying a certain length of the pipe. Then if the number of pounds to be delivered per minute is known (such as the pounds used per minute by an engine) the length of pipe to contain one minute's supply will be the two multiplied together. Therefore, the pound of steam just entering the pipes will have to travel that distance in a minute. This will give the rate of flow regardless of the length of pipe in actual use. Take an example:

"To ascertain the velocity of the flow of steam or other fluids in pipes for a given output in pounds: Find the number of feet in length of a pipe of the size given (actual inside measurement) to contain one pound of steam at the given pressure or density (see steam tables); multiply this length by the number of pounds delivered per unit of time (minute or second), and the product will be the velocity necessary for that amount of steam.

"*Example*—If 100 lb. of steam at 100 lb. absolute pressure is flowing through a 4-in. pipe, what is its velocity? The nominal area of a 4-in. pipe = 12.73 sq.in. (1 ft. in length = 153 cu.in.) and 11.3 ft. in length = 1 cu.ft. One pound of steam at the given pressure occupies 4.43 cu.ft., therefore 1 lb. of steam will occupy $4.43 \times 11.3 = 50$ ft. in length of the pipe. Then 50 ft. $\times$ 100 lb. = 5,000 ft. per minute velocity. At 153 lb. pressure the volume of 1 lb. of steam is 3 cu.ft., therefore, $3 \times 11.3 = 34.0$ ft. in length of 4-in. pipe to contain 1 lb. of steam, and then for the same weight (100 lb.) delivered per minute the velocity will be $34 \times 100 = 3{,}400$ ft. per minute.

"To find the quantity of steam or other fluid which will be delivered when the velocity and pipe size are known: Ascertain the number of feet in length of pipe to contain one pound of steam at the given pressure or density, as before; divide the known velocity per minute by this and the quotient will be the number of pounds delivered per unit of time.

"To find the size of pipe necessary to deliver a given quantity (pounds) of steam or other fluid at a predetermined or desired velocity: Divide the velocity per minute by the quantity (pounds) to be delivered, and the quotient will be the number of feet in length of the pipe to contain one pound of steam. This must be reduced into inches in diameter (the reverse of the opera-

tion in the first problem) by first dividing this quotient by the volume in cubic feet of 1 lb. of steam at the given pressure, or density. This gives the number of feet in length of pipe equal to 1 cu.ft. of space. This, divided into 1,728 cu.in. per ft., gives the square-inch section of the pipe. See the diameter of the pipe nearest to this area.

"I have arranged for my own convenience a table of pipe dimensions which I use most frequently and find it more convenient than looking up several detached tables. For any such problems you need only the steam tables and this one."

TABLE OF THE PRINCIPAL DIMENSIONS OF PIPES

Nominal Size, In.	Threads per In.	Weight per Ft.	Thickness of Metal, In.	Diameter, External, In.	Circumference, External, In.	Area, External, Sq.In.	Length in Ft. = 1 Sq.Ft., External	Diameter, Internal, In.	Circumference, Internal, In.	Area, Internal, Sq.In.	Length in Ft. = 1 Sq.Ft., Internal	Length in Ft. = 1 Cu.Ft. Capacity
⅛	27	0.244	0.068	0.405	1.272	0.129	9.431	0.270	0.848	0.0573	14.199	2513.300
¼	18	0.424	0.088	0.540	1.696	0.229	7.073	0.364	1.144	0.1041	10.493	1383.300
⅜	18	0.567	0.091	0.675	2.121	0.358	5.658	0.494	1.552	0.1917	7.747	751.200
½	14	0.850	0.109	0.840	2.639	0.554	4.547	0.623	1.957	0.3048	6.141	472.400
¾	14	1.130	0.113	1.050	3.299	0.866	3.637	0.824	2.589	0.5333	4.635	270.250
1	11½	1.678	0.133	1.315	4.131	1.358	2.904	1.048	3.292	0.8626	3.641	166.900
1¼	11½	2.272	0.140	1.660	5.215	2.164	2.301	1.380	4.335	1.496	2.767	96.250
1½	11½	2.717	0.145	1.900	5.969	2.835	2.010	1.611	5.061	2.038	2.372	70.660
2	11½	3.652	0.154	2.375	7.461	4.430	1.608	2.067	6.494	3.356	1.847	42.910
2½	8	5.763	0.203	2.875	9.032	6.492	1.328	2.468	7.753	4.784	1.547	30.100
3	8	7.575	0.216	3.500	10.996	9.621	1.091	3.068	9.638	7.388	1.245	19.500
3½	8	9.109	0.226	4.000	12.566	12.556	0.954	3.548	11.146	9.887	1.076	14.570
4	8	10.790	0.237	4.500	14.137	15.904	0.848	4.026	12.648	12.730	0.948	11.310
4½	8	12.538	0.247	5.000	15.708	19.635	0.763	4.508	14.162	15.961	0.847	9.020
5	8	14.617	0.258	5.563	17.477	24.306	0.686	5.045	15.849	19.990	0.756	7.200
6	8	18.974	0.280	6.625	20.813	34.472	0.576	6.065	19.054	28.890	0.629	4.980
7	8	23.544	0.301	7.625	23.955	45.664	0.500	7.023	22.063	38.738	0.543	3.720
8	8	25.000	0.322	8.625	27.096	58.426	0.442	7.982	25.076	50.040	0.473	2.880
9	8	34.188	0.342	9.625	30.238	72.760	0.396	8.937	30.238	62.730	0.427	2.290
10	8	35.000	0.365	10.750	33.772	90.763	0.355	10.020	31.476	78.839	0.375	1.820
11	8	45.557	0.375	11.750	36.914	108.430	0.325	11.000	34.557	95.033	0.347	1.515
12	8	49.562	0.375	12.750	40.055	127.677	0.300	12.000	37.700	113.098	0.319	1.270
13	8	54.568	0.375	14.000	43.982	153.938	0.272	13.250	41.626	137.887	0.288	1.044
14	8	58.573	0.375	15.000	47.124	176.715	0.254	14.250	44.768	159.485	0.268	0.903
15	8	62.579	0.375	16.000	50.265	201.062	0.239	15.250	47.909	182.655	0.250	0.788

Operating Transformers on Different Frequencies

By Gordon Fox

Transformers are designed and rated for specified voltages, but by means of taps, connections and combinations much flexibility in application is possible. Consideration of the many permissible connections is not within the province of this discussion, which will take up only the relations existing when transformers designed for a given frequency are operated upon lines of another frequency.

The formula showing the relations between the different factors of transformer design and action, is as follows:

$$V = 4.44\ A\ B\ N\ f \div 10^8$$

Where,

V = Applied voltage;
A = Magnetic cross-section, in square inches;
B = Lines of force per square inch;
N = Number of turns in series in winding;
f = Frequency.

Consider a 60-cycle transformer that is to be used upon a 25-cycle circuit. The frequency is here a little less than half its intended value. It can be seen from the formula that the magnetic intensity is inversely proportional to the frequency of the system. The reduced frequency will thus greatly increase the magnetism in the transformer iron.

This increase in magnetic intensity greatly increases the iron losses. The transformer iron therefore becomes overheated, and the winding is likely to burn out if kept in service continuously. If the 60-cycle transformer is to be operated at 25 cycles, it is necessary that the magnetic density be reduced in some manner. The formula shows that this varies directly with the voltage. It is therefore possible to reduce the magnetic density by reducing the impressed voltage. If the voltage at 25 cycles be one-half of that for which the transformer is rated at 60 cycles, the decrease in voltage will just about compensate for the decrease in frequency and the induction will be nearly the same as at 60 cycles, full voltage. With a fixed connection the transformation ratio is the same at any frequency, the voltages being always approximately proportional to the number of turns. Thus a 2200-1100 to 220-110-volt, 60-cycle transformer would be suited for 1100-550 to 110-55 volts if used on 25 cycles.

The kv.-a. rating of a transformer is the product of kilovolts and amperes. The transformer windings will carry the same current at 25 cycles and 60 cycles with about the same degree of heating. But the voltage was reduced one-half when the 60-cycle transformer was connected to a 25-cycle circuit. Therefore the kv.-a. capacity is correspondingly reduced; that is, a 60-cycle transformer will develop about half its rated capacity when connected to a 25-cycle circuit. This same line of reasoning shows why it is that 25-cycle transformers are larger and heavier than 60-cycle transformers of the same kv.-a. ratings.

It is both possible and feasible in many cases to utilize 25-cycle transformers upon 60-cycle circuits. They will then operate at low magnetic densities and will deliver an amount of 60-cycle power corresponding to the 25-cycle ratings. In order to utilize the full capacity of the transformer it may be operated upon a 60-cycle circuit of double the voltage of the 25-cycle rating, provided, of course, that the insulation will stand such a change. Most transformers are tested at the factory under double voltage without oil, and a sufficient factor of safety is involved to make this practice permissible. A point that enters here, however, is the eddy currents within the conductors themselves. At 60 cycles these will probably be higher than at 25 cycles, so that the regulation will be a little poorer and the copper losses greater at the higher frequency. The allowable rating may be thereby slightly reduced.

Hard Lead is lead containing antimony. An important effect of antimony in lead is to cause the alloy to expand when cooling from the molten state.

The Two-Stage Condenser

By Paul Bancel

SYNOPSIS—The author proposes a two-stage condenser, in the first stage of which part of the steam is condensed at a lower vacuum, the rest passing through the final stage of the turbine and being condensed in the other portion of the condenser at a higher vacuum. The flows of circulating water, condensate and air are all countercurrent. The advantages of the design are discussed for installations where a natural supply of cold condensing water is available, also where the water must be artificially recooled.

Further utilization of the energy of steam made available by expansion to high vacuums still presents attractive possibilities for increased economy of steam turbines. With original conditions of 200-lb. pressure and 150-deg. superheat the gain by expansion from 29-in. vacuum to 29¾-in. vacuum is about 60 B.t.u. of available energy, or 15 per cent. (see lines 3 and 7 of the following table). For each degree lowering of the exhaust temperature the gain in energy is roughly 1.55 B.t.u., as shown in the last column.

Exhaust Steam, Temp. F.	Vacuum, Inches Mercury Referred to 30-In. Barometer	Energy Available for Conversion into Work, B.t.u. per lb.	Difference per Deg. of Exhaust Temp., B.t.u.
100	28.07	369.6	....
90	28.58	384.6	1.50
80	28.97	400.0	1.54
70	29.26	415.6	1.56
60	29.48	430.9	1.53
50	29.64	447.3	1.64
40	29.75	463.4	1.61
Average			1.56

The range of vacuums from 29 to 29¾ in. corresponds approximately to temperatures from 80 to 40 deg., which are still somewhat above available water temperatures, summer and winter. Thus there are no temperature limitations, at least, to the attainment of these vacuums. The commercial limitations hinge upon the extent of the difference allowed between the incoming condensing water and the temperature of the exhaust steam. There are two heat steps which make up the total overall temperature difference—the rise in temperature of the water as it takes up its heat and the final difference between the temperature of the outlet circulating water and the steam temperature which must always exist unless infinite condensing surface is used.

By increasing the amount of condensing water, the discharge-water temperature can be reduced until, when an infinite amount of water is used, its outlet is the same as its inlet temperature. Similarly, by increasing the amount of surface, the spacing of the tubes and capacity of the air pump, the vacuum temperature can be brought down closer and closer to the discharge circulating water temperature, approaching the condition of zero "outlet temperature difference" or of a vacuum temperature equal to the discharge circulating-water temperature.

As approximating these limits of zero water rise and zero outlet difference, let us consider an installation using sufficient water to give a 5-deg. temperature rise with a condenser having the surface, tube spacing and air-pump capacity to give a 5-deg. outlet temperature difference. The exhaust-steam temperature would then be only 10 deg. higher than the entering cooling-water temperature, which would mean a vacuum of 29.7 in. in winter with 32-deg. water, a vacuum of 29 in. in summer with 70-deg. water, and a mean vacuum for the year of about 29½ in.

This assumes water obtained from a large source in a temperate climate. A curve of water temperatures for a year shows that the warm-water periods are of short duration, while for eight or nine months the temperature is well below 60 deg. The average weighted by months for Lake Michigan is 47 deg. for example.

To obtain a 5-deg. rise in circulating water, about 190 lb. of water per pound of steam would be needed, corresponding to 85,000 gal. per min. for a 20,000-kw. installation. The cost of pumping the water would be in proportion—that is, two to three times as great as that found in recent installations—and would offset in part the gain from the higher vacuum.

The cost of the power consumed by the auxiliaries depends but slightly on the system of drive. If noncondensing turbines exhausting to the feed heater are used, the heat retrieved is available to reduce the coal consumption, but should not be credited solely to the auxiliary power but to the whole plant output. For suppose that the auxiliaries were driven directly or electrically and the feed heated by bleeding steam from the main turbine; one would not credit the heat retrieved in that case to the auxiliaries but to the total output. That it is becoming generally appreciated that there is no inherent saving by using steam-driven auxiliaries to supply heat for the feed is evidenced by the rapidly growing use of motor drive. Several plants have recently installed a system involving an independent feed-heating turbine driving a generator which supplies power for motor-driven auxiliaries and exhausts to a heater or condenser supplied with the boiler feed as injection water. The feed-heating turbine is tied in electrically with the main unit and can either generate additional power over and above that needed for the auxiliaries or else call upon the main turbine for extra power to make up its shortage. The system being elastic, permits of adjustment of load to obtain any desired feed temperature for delivery to economizers or boilers. The best feed temperature is that coinciding with a minimum loss in heat to the stack and in the condensing-water discharge.

This arrangement approaches the ideal, since steam can be withdrawn from the thermal system at any pressure and in any quantity, as contrasted to the fixed quantity and pressure where auxiliary turbines are used. The power required for the auxiliaries, however, still represents so many kilowatts, no matter what means are used for feed heating. Thus in striving for high vacuums, as by the use of enough circulating water to give only 5-deg. rise, as in the present example, one must not lose sight of the loss in the auxiliary power.

Turning to the condenser, which would have to have sufficient surface to maintain the vacuum temperature within 5 deg. of the outlet-water temperature, the tem-

perature "head," or difference between vacuum temperature and mean water temperature, to cause heat transmission would be 7½ deg., so that the heat per square foot of surface per hour, on a basis of 400 B.t.u. per degree of difference, would be 3000, and approximately 3.5 sq.ft. of surface would be needed per kilowatt.

Besides the large amount of surface used there would have to be wide and generous spacing of the tubes in order to insure penetration of the stream into the condenser at the very high vacuum contemplated. As against present design, the spacing would have to be two or three times as wide in order to maintain the pressure drop within a limit of 0.1 in. to 0.2 in.

This pressure difference could not be maintained with an ordinary pump under the conditions of very high vacuum. If we assume a hurling-water air pump using 70-deg. water, the theoretical 100 per cent. vacuum is 29.26 in., which would fall to 99 per cent., or about 29 in., when handling its load of air and intermixed vapor, and even lower if the hurling water is recooled distilled water. But the 29-in. vacuum is desired at the turbine with 70 deg. water, in the present example, and thus there would remain no margin for the drop through the condenser.

The solution would be to use a "booster"—in other words, a steam jet in series with the air pump, and it is known that under these conditions the steam jet maintains a pressure of 0.1 in. or less absolute, or 29.9 in. vacuum, and there would then be a large permissible drop through the condenser. The steam jet, or "augmentor," also improves the rate of heat transmission, due to the removal of the air in a more rarefied state, provided the pressure drop is not excessive.

These refinements in design and the additional fixed and operating charges on the condensing equipment entailed in order to obtain very high vacuums would be warranted only so long as the turbine could profitably utilize the energy made available.

Contrary to prevalent opinion, the steam turbine cannot utilize the highest attainable vacuums unless specially designed for that purpose. In writing on this subject the editor of *Engineering*, London, quotes figures published by M. G. L. Swallow as the results of careful tests on a large and highly efficient turbine: "On increasing the vacuum from 27 to 28 in. the actual reduction in the steam taken per kilowatt hour was 5 per cent., while that theoretically due was 6.8 per cent. On still further raising the vacuum to 29 in., the actual gain was 6 per cent., which must be compared with the theoretical figure of 11.4 per cent." [1]

In a paper in the same issue of *Engineering* Mr. Stoney cited as an example a turbine of 3000 kw. at 3000 revolutions (corresponding approximately to 2000 kw. at 3600 revolutions for American practice) in which the actual gain from 28 to 29 in. of vacuum was only 6½ per cent., there being a leaving loss of 12 B.t.u. per lb. "For still higher powers, however, these effects become more pronounced, until conditions are eventually reached in this class of turbine having a highly restricted exhaust end, when an increase in vacuum causes no gain."

The small gain from increase of vacuum at large capacities is shown by tests of a 10,000-kw. double-flow turbine, reported by Sam L. Naphtaly, "Transactions," American Society of Mechanical Engineers, Vol. 32. It was found that one inch of vacuum affected the steam consumption only 3 per cent. at full load as against 6 per cent. at half load.

Let us consider the conditions existing at the last set of blades, from which the steam issues at high vacuum. Irrespective of the rotation of the blades and of the velocity of the steam relative to the blades, there is a flow in an axial direction which, together with the area through the annulus formed by the blades, determines the steam volume issuing from the turbine. All of the steam flowing into the turbine at one end must come out through this annulus at the other. As the area for flow is fixed by the mean perimeter and blade height and the space occupied by the blades, it follows that the residual velocity of the steam issuing from the last stage depends directly on its volume or the weight times the specific volume. This velocity is not to be confused with the velocity of the steam through the exhaust nozzle of a horizontal turbine, into the condenser, which must be created by further conversion of energy. The residual velocity here considered is that parallel to the axis of the turbine.

The energy rejected in this manner varies as the square of the velocity and amounts, for example, to 8.6 B.t.u. per lb. for 657 ft. velocity and 32 B.t.u. per lb. for 1269 ft. velocity, these velocities having been selected as equal numerically to the specific volumes of steam at 29 and 29½ vacuum respectively.

Thus a turbine having a leaving loss of 8.6 B.t.u. at 29-in. vacuum has a leaving loss of 32 B.t.u. at 29½ in. vacuum, assuming that the total weight of steam flowing is the same in both cases. The increase in vacuum results in 32—8.6, or 23.4 B.t.u. additional leaving loss. Now the gain in energy by expanding to 29½-in. vacuum as against 29 in., from the same original conditions, is about 30 B.t.u.; so that the net gain would be only 30 — 23.4 = 6.6 B.t.u., or on a basis of 400 B.t.u. available at 29 in., 1.65 per cent. net gain. This explains why the actual gain as compared to the theoretical falls off rapidly as higher and higher vacuums are maintained. If the turbine is designed for a 26-in. vacuum these conditions are met with at about 28 in.; if designed for 28 in., they become serious around 29 in. and if designed for 29 in. they become serious around 29½ in.

From the 1.65 per cent. net gain obtained in the foregoing example there must be subtracted the loss of 20 deg. in condensate temperature, or about 1⅔ per cent., which wipes out the entire gain due to higher vacuum, to say nothing of the increased power for the auxiliaries.

All the refinements and additional expenditures in equipment and power, in order to obtain the higher vacuum are thus a complete loss unless the turbine is designed to utilize properly the energy made available. Greater area must be allowed in the final stages—for example, by decreasing the rotative speed of the turbine and thus making the wheel perimeter larger, or by decreasing the load and weight of steam flowing, or by changing a single-flow turbine to a double-flow, or a double-flow to a quadruple-flow. The leaving velocity would be cut in half by the last two changes—that is, from 1269 to 635, which is even slightly less than the 657-ft. velocity for the 29-in. vacuum turbine. The gain in energy

[1] "Engineering," Nov. 27, 1914.

by expansion to the higher vacuum would then be equal to the theoretical, and the turbine economy would be increased 7½ per cent.

This increase in economy would be profitable only so long as it overbalanced the added cost of turbine and condenser, the power expenditure for obtaining the vacuum and the loss of heat in the colder condensate.

The Two-Stage Condenser

Suppose that instead of making a change in the turbine speed, in the capacity, or in the number of final stages in parallel, the whole of the steam is expanded down to 29 in. in the same turbine, and then only half of it is further expanded to 29½ in. through an additional stage of the same size as the last. No more energy for conversion into work is then made available than can be profitably utilized in the same-sized final stage.

From the condenser standpoint, only half the steam is stepped down to the lower temperature, giving up its heat to the circulating water in the first half of its rise, while the remainder of the steam gives up its heat to the circulating water in the second half of its rise. The steam exhausted from the turbine at the high vacuum is condensed in one section of the condenser (the upper portion in Fig. 1), and the other half, bled out at 29 in., is condensed in a second section.

Having a large volume of comparatively hot steam at the lower vacuum, the cold condensate from the high-vacuum condenser is passed into the low-vacuum section through a suitable trap *A* and allowed to drip over a heater tray *B*, so that its temperature is raised, and in this way heat which would otherwise be rejected to the circulating water is retrieved.

The air from the high-vacuum section is discharged into the low-vacuum condensing space by steam jets, one of which is shown at *C*, and finally withdrawn along with the air from this section by the customary air pump. The air outlet nozzles for connection to the air pump are not shown in section of Fig. 1.

The refinements of extra-wide tube spacing and steam jets are required in the high-vacuum condensing section only. The efficiency of heat transfer is augmented by reason of the wedge shape of the condensing chamber, which may be assumed to consist of a number of pie-shaped sectors and which thus approximate the theoretical shape for the elimination of dead corners and air pockets; and, secondly, by the use of multiple steam jets arranged along the length of the condenser shell, thus insuring distribution of steam and equal activity of the surface along the entire length of the tubes.

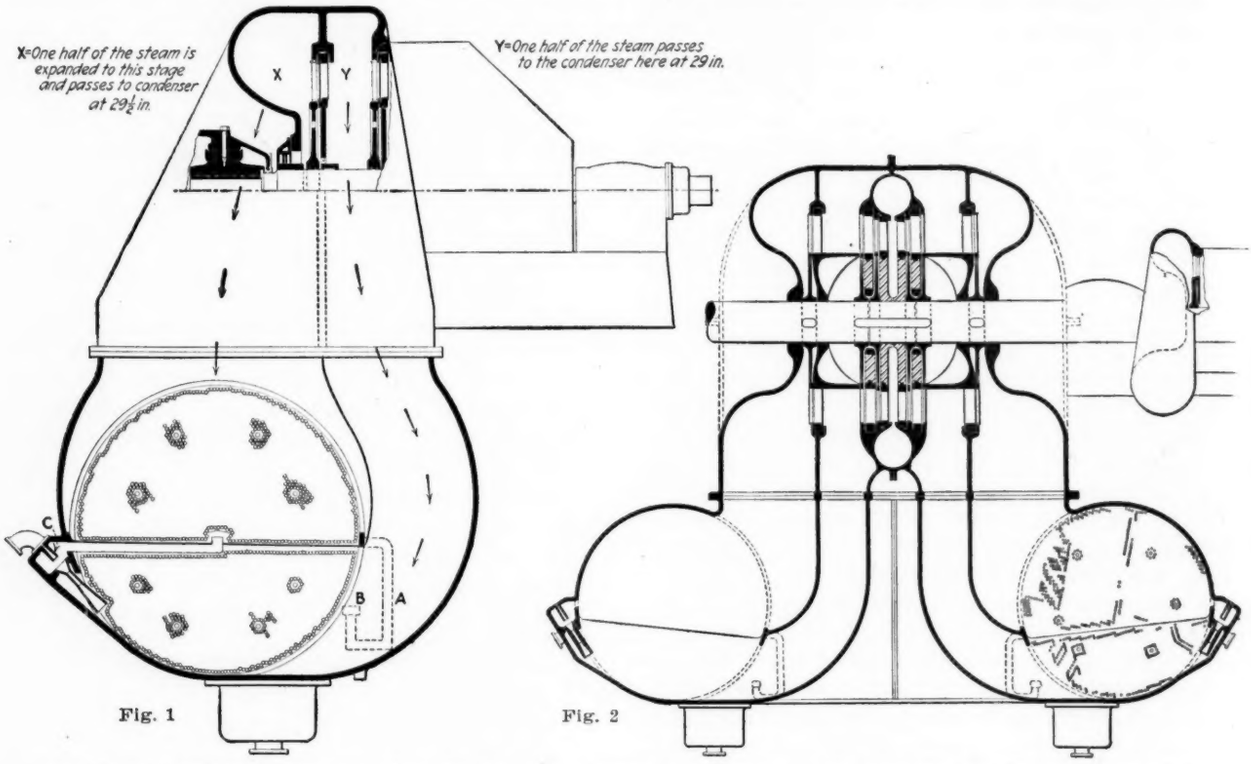

X=One half of the steam is expanded to this stage and passes to condenser at 29½ in.
Y=One half of the steam passes to the condenser here at 29 in.
X
Y
C
B
A

Fig. 1 Fig. 2

SHOWING APPLICATIONS OF TWO-STAGE CONDENSER TO SINGLE- AND DOUBLE-DISCHARGE TURBINES

For large capacities and high vacuums the entrance passage for the high-vacuum steam alone may be equal to the full length and diameter of the shell, in which case the condenser would be placed directly under the high-vacuum exhaust chamber. This design allows all the customary flow area for only half the steam, which may therefore have twice the specific volume, and furthermore by the arrangement shown in Fig. 1 the steam is distributed all around the periphery of the tube bank and the air is withdrawn from the center, so that there is a minimum loss in pressure through the shorter tube bank. The same design can also be applied to the low-vacuum section.

The circulating water passes through the tubes of the high-vacuum section first, and there is a small rise in temperature and low discharge temperature. But the quantity of water used is not excessive, because the same water condenses the other half of the steam at the low vacuum. The water is thus used twice.

In many plants the water supply is limited or has to be pumped a great distance or artificially recooled, so

that the operating and fixed charges per pound of cold circulating water delivered to the condenser are high, making it very desirable to use a small quantity. This applies particularly to cooling-tower installations, which will be treated later.

When using a natural supply of water it is obvious that extremely high vacuums can be maintained in the high-vacuum section of the two-stage condenser, and that the energy made available can be profitably utilized. The percentage of steam drawn through the last wheel or stage is determined by the fixed water conditions and the amount of surface used in each section. Manipulation of the steam jets permits delicate adjustment in the ratio desired.

Table 1 summarizes the comparative economy of various combinations for 20,000 kw. at best load. An installation for an average yearly vacuum of 29 in. is given in the first column. The turbine design is such as to give only 7 B.t.u. leaving loss at 20,000-kw. load, which is very low and corresponds to a steam velocity of about 590 ft. per sec. The equipments and comparative economies are then tabulated for (Case 2) 29½-in. vacuum, (Case 3) 29 and 29½ with a two-stage condenser, and (Case 4) 29.28-in. on all the steam (corresponding to 29 in. with 57-deg. water) which is about the standard now in use.

TABLE I. TURBINE AND CONDENSER FOR 20,000-KW. (BEST LOAD) AND HIGH VACUUM

	Case 1 (29-In. Vacuum with 47-Deg. Water)	Case 2 (29½-in. Vacuum with 47 Deg. Water)	Case 3 (Two-Stage Condenser; Vacuums 29 and 29½ In. with 47 Deg. Water)	Case 4 (29.28-In. Vacuum on all Steam with 47 Deg. Water; Corresponds to 29-In. with 57 Deg. Water)
Condenser for 20,000 kw. (12 lb. per kw.-hr., 950 B.t.u. per lb., 400 B.t.u. coefficient).	21,000 Sq.Ft.	50,000 Sq.Ft.	36,000 Sq.Ft.	31,000 Sq.Ft.
Quantity of Circulating water and power required.	10-deg. rise, 43,000 g.p.m., 360 B.hp.	5-deg. rise, 85,000 g.p.m., 715 B.hp.	10-deg. rise, 43,000 g.p.m., 360 B.hp.	10-deg. rise, 43,000 g.p.m., 360 B.hp.
Air pump.	Rotary or dry vacuum, about 100 hp.	Same with addition of steam gets using ⅓% of steam.	Same with steam gets for high vacuum, using ¼% steam.	Somewhat larger pump on account of high vacuum compared to Case 1.
Maximum condensate temperature.	79°.	60°.	79°.	69°.
Turbine.	Single-flow at 1200.	Double-flow at 1200.	Single-flow at 1200.	Single-flow at 1200.
Theoretical energy from initial condition of 200 lb. and 150 deg. superheat.	402 B.t.u.	431 B.t.u.	431 402	417
Leaving loss.	7 B.t.u.	6 B.t.u.	6 B.t.u. 7 B.t.u.	13 B.t.u.
Net energy.	395	425	425 395 } 410	404
Gain compared to 29-in.	.....	7½%.	3½%.	2¼%.
Loss in steam jets or air pump.	.....	—½%.	—¼%.	—.1.
Loss in circulating pump power.	.....	—1.7%.	.....	.....
Loss in colder condensate.	.....	—1.6% (19°).	.....	0.85% (10°).
Net power gain compared to Case 1.	.....	3.7%.	3¼%.	1.30%.
Additional equipment compared to 29-in., Case 1.		29,000 sq.ft. more condenser surface; pump, piping and tunnels for 100% more water; 100% more final wheel area, requiring double flow at 1200 revolutions.	15,000 sq.ft. more condenser surface; single-flow turbine, with additional wheel of same size as last wheel.	10,000 sq.ft. more condenser surface and larger air pump; turbine same as Case 1.
Net power gain compared to Case 4.	.....	2.4%.	2.2%.	.....
Additional equipment compared to Case 4.	.....	19,000 sq.ft. more condenser surface; pumps, piping and tunnels for 100% more water; 100% more final wheel area of turbine requiring double flow at 1200.	5000 sq.ft. more surface; turbine same size, single-flow with additional last wheel.	

Compared to the last case—that is, the 29.28-in. vacuum with 47-deg. water—there is a gain of 2.2 per cent. by operating at vacuums of 29 and 29½ in. in a two-stage condenser, requiring a turbine of the same size as is customarily used, with an additional final stage, the same size pump, piping, tunnels, etc., and only 5000 sq.ft. more surface. This assumes no reduction in surface due to the use of the steam jets. Taking the additional cost of the two-stage condenser and double outlet turbine at $7000, the annual charge at 12 per cent. is $840. The saving of power is 2.2 per cent. on 20,000 kw. × 4000 hrs. = 1,760,000 kw.-hr., which is worth $4400 at ¼c. per kw.-hr.

This is to be contrasted with (Case 2), which gives a net gain of only 2.4 per cent., using a turbine of double the area in the final stages for 29½-in. vacuum on all of the steam, a condenser of 19,000 sq.ft. more surface, and pumps, piping, tunnels, etc., for twice the quantity of water.

Application to Large Units

The two-stage condenser design permits the installation of large units on limited floor space in locations where the cooling-water supply is becoming limited and in place of older units of smaller capacity. In order to reduce the size and weight of both turbine and condenser, the turbine may be operated at somewhat lower vacuums, 28 and 29 in., for example. As only half the steam is expanded to 29 in., it is possible to double the capacity of a turbine for the same area in the final stage, and as the other half of the steam is condensed at a lower vacuum, double the capacity is also obtained using the same quantity of water. The surface also is greatly reduced, with corresponding reduction in cost and bulk of the entire equipment. The loss in efficiency due to bleeding out half the steam at 28 in. is offset by the savings in circulating-pump power, air-pump power, hotter condensate and reduced fixed charges on the plant. It is also to be remembered that by increasing the capacity of large units by this design there is a gain in efficiency due to the saving of power in bearing friction. Fig. 2 shows how a double-flow turbine for, say, 30,000 kw. at 1500 revolutions and 29-in. vacuum, may be arranged for two-stage condensation at vacuums of 28 and 29 for 60,000 kw.; vacuums of 28.7 and 29.3 for 45,000 kw.; vacuums of 29 and 29.5 for 30,000 kw.

Application to High-Speed Turbines

High turbine speeds mean small wheel diameters and low friction losses, particularly in the high-pressure stages; but the higher the speed, the less the area of the last stages and smaller the capacity of the turbine. Thus turbines for 3600 to 5000 and 6000 revolutions and higher can be built only for small capacities if high vacuums are to be utilized.

The following assumptions are made for calculating the capacities given in the following table.

A mean peripheral speed of 600 ft. per sec.

A maximum blade height of one-fifth the wheel diameter.

A leaving velocity of 590 ft. per sec., corresponding to 7 B.t.u. loss.

A steam consumption of 15 lb. per kw.-hr. at 28-in. vacuum.

A thickness coefficient for the blades offsetting the decrease in volume of the steam due to condensation.

Vacuum	5000 Rev. Kw.	3600 Rev. Kw.
28	1380	2670
28½	1084	2100
29	780	1510
29¼	620	1200
29½	490	950

The capacities for 5000 revolutions are just about half those for 3600 revolutions, and both decrease rapidly with increase in vacuum. These ouputs are cut approximately in half for low-pressure turbines—that is, for 29-in. vacuum at 5000 revolutions the capacity is about 390 kw. and at 3600 revolutions about 750 kw.

There is a large discrepancy between the capacities of these turbines and the driven machines. At 3600 revolutions alternating-current generators are built for 4000 kw. or more, which is almost three times the power of the turbine at 29-in. vacuum high pressure and over

TABLE 2. TURBINE AND CONDENSER FOR 1,500 KW. AT HIGH VACUUM AND HIGH SPEEDS

	Case 1 Turbine, 3,600 revolutions at 29-in. vacuum.	Case 2 Turbine, 5,000 revolutions at 28 and 29-in., two-stage.
Condenser for 21,000 lb. steam; coefficients, 350 and 600 with ejectors.	3,500 sq.ft.	2,000 sq.ft.
Quantity of circulating water and power.	10° rise; 4,000 g.p.m.; 50 kw.	20° rise; 2,000 g.p.m.; 25 kw.
Air pump.	Dry vacuum pump for 29-in. vacuum, 15 kw.	Dry vacuum pump for 28-in. vacuum, 7.5 kw.
Maximum condensate temperature.	79°.	101°.
Heat available from 200 lb. and 150° superheat.	402 B.t.u.	402 370 B.t.u.
Per cent. loss in turbine power.	.....	—4%.
Loss in steam jets in per. cent. turbine power.	.....	—½%.
Gain in air pump power, in per cent. turbine power.	.....	+½%.
Gain in circulating pump power, in per cent. turbine power.	.....	+1½%.
Gain in condensate heat, in per cent. turbine power.	.....	2%.
Net loss in economy in per cent. turbine power.	.....	—½%.
Equipment.	Turbine, 3,600 revolutions; condenser, 3,500 sq.ft.; pump, 4,000 g.p.m. and large air pump with driving motors of corresponding size.	Turbine, 5,000 revolutions with double outlet for two-stage condenser; condenser, 2,000 sq.ft.; pumps, ½ size.

five times the capacity of a turbine at 29-in. vacuum low pressure.

Similarly, it is possible to build reduction gears of large capacities for 5000 revolutions to drive pumps or generators, whereas the power at 29-in. vacuum is limited to 780 kw. high pressure and 390 kw. low pressure.

As an example of how the two-stage condenser permits of extending the capacity of a turbine without reduction in speed, let us assume that a 1500-kw. turbine at 29-in. vacuum is desired. According to the table it operates at 3600 revolutions. Suppose, however, that the turbine is of the double-vacuum design for a two-stage condenser and that only half the steam is exhausted at 29 in. So far as concerns the last stage, then, the rating is 1500 ÷ 2, or 750 kw., and the 5000-revolution machine can be used, the table giving 780 kw. as the capacity. The capacity at vacuums of 28 and 29 in. thus becomes 2 × 780, or 1560 kw.

The bulk and cost of the turbine are reduced in approximately the inverse ratio of the squares of the speeds, or about one-half. There is a loss in turbine economy of 4 per cent., due to the bleeding out of steam at 28-in. vacuum; but this is offset by the lower steam friction losses with smaller wheels at high speed, by reduced bearing loss due to the lighter rotor, by the fact that the condensate is 22 deg. hotter, or a gain of 2 per cent.,

TABLE 3. TURBINE, CONDENSER AND COOLING TOWER FOR 1000 KW. ATMOSPHERIC CONDITIONS, 70 DEG. AND 70 PER CENT. HUMIDITY

	Case 1 Ordinary Equipment for 27-In. Vacuum)	Case 2 Two-Stage Condenser for Vacuum of 27 and 28.1-In.	Case 3 Two-Stage Condenser for Vacuum of 25½ and 27½-In.	Case 4 All Steam at 27.6-in. (Mean of 27 and 28.1 In.)
Condenser for 1000 kw.; 20,000 lb. steam; coefficients, 300 in standard condenser and 600 where steam jets are used.	3,300 sq.ft.	4,000 sq.ft.	2,500 sq.ft.	4,100 sq.ft.
Quantity of circulating water and power.	1,200 g. p. m., 30 B.hp.	1,200 g.p.m., 30 B.hp.	900 g.p.m., 23 B.hp.	1,600 g. p. m., 40 B.hp.
Air pump.	Reciprocating wet vacuum pump of "valveless" type.	Same as Case 1.	Cheap form of pump for 25½-in. vacuum.	Reciprocating wet vacuum pump larger than Case 1.
Maximum condensate temperature.	115°.	115°.	130°.	107.5°.
Tower.	Capacity, 1,200 g.p.m. from 110 to 80 deg.	Same size as Case 1.	900 g. p. m. from 120 deg. to 80 deg. (33% smaller).	Tower to cool 1,600 g. p. m. from 102.5 to 80 deg. (35% larger).
Fan power.	25 B.hp.	25 B.hp.	17 B.hp.	34 B.hp.
Turbine.....	Standard 27-in. vacuum turbine.	Same with additional wheel of same size as last wheel.	Standard 25½-in. vacuum turbine with additional wheel of same size as last.	For same leaving loss, turbine must be 23% larger.
Heat available to turbine.	Heat available from 150 lb. to 27-in. vacuum = 304.	Heat available to 27 in. vacuum = 304, to 28.1-in. vacuum = 325.	Heat available to 25½-in. vacuum = 282, to 27½-in. vacuum = 313.	Heat available to 27.6-in. vacuum = 314.
Per cent. gain or loss in power in per cent. of turbine power.	.....	+3.45%.	—2.1%.	+3.33%.
Loss in steam jets or air-pump power in per cent. of turbine power.	.....	—.25%.	—.25% steam jets; +.25% air pump.	—.25% air pump.
Saving or loss in circulating power in per cent. of turbine power.	.....	.....	+.65%.	—.93%.
Saving or loss in fan power in per cent. of turbine power.	.....	.....	+.70%.	—.84%.
Saving or loss in condensate heat in per cent. turbine power.	.....	.....	+1.40%.	—0.70%.
Net gain in economy in per cent. turbine power.	.....	+3.2%.	+.65%.	+.61%.
Equipment.	.....	Turbine has additional wheel; condenser, 700 sq. ft. more surface; cooling tower, pump and piping remain same size.	Turbine of smaller size (¼ smaller final wheel area), with additional wheel; condenser, 800 sq.ft. less; pump and piping, 25% smaller; tower, 33% smaller.	Turbine, 23% larger; condenser, 800 sq.ft. more; circulating pump and piping, 33% larger; tower, 35% larger.

and a saving of over 2 per cent. in the power for the auxiliaries. At the same time the cost of turbine, condenser, piping, auxiliaries and power-house construction is decreased. The surface required in the condenser is decreased from 3500 to 2000 sq.ft. and the amount of cooling water from 4000 gal. to 2000 gal. per min. These figures are tabulated in Table 2.

The possibilities of obtaining very high vacuum with small condensing surface in a unit of this size are perhaps better than that with very large condensers. While the proportionate air leakage may be slightly higher, the

small depth of the tube bank and the comparatively few rows of tubes result in small pressure drop through the condenser. Using distributed steam ejectors for exhausting the air and with the wedge-shaped condensing chambers and proper tube spacing, rates of heat transmission approximating 1000 B.t.u. per square foot of surface per hour per degree difference are obtainable.

Advantages in Cooling Tower Installations

In cooling tower installations, lower discharge-water temperatures, to give higher vacuums, require not only the pumping of more cooling water but a large increase in the cost of the cooling tower and the power for its operation. At the same time, the head against which the water must be pumped is larger than in the customary installation on account of the elevation of the distributers in the tower. For these reasons the condensation of the steam at two vacuums in a two-stage condenser, with water heated to the final temperature in two steps, produces a gain in economy which cannot be obtained even by the expenditure of far more money for a large equipment of the standard design.

A cooling tower and condensing installation are usually designed for 27-in. vacuum, with water rising in the condenser from 80 to 110 deg., and cooled in the tower from 110 to 80 deg., with atmosphere about 70 deg. F. and 70 per cent. humidity (Table 3, Case 1).

The two-stage condenser operating with the same cooling tower heats the water in two parts from 80 to 95 deg. and from 95 to 110 deg. and maintains vacuums of 28.1 and 27 in., with an increase in economy of 3.2 per cent. for a very slight additional fixed charge (see Table 3, Case 2).

In comparison Case 4 gives the equipment and economy obtained when operating the turbine at the mean of these vacuums, which is 27.6 in. The turbine must be 25 per cent. larger at its last stage, 800 sq.ft. more condensing surface is needed, the pump and piping must be larger, and the cooling tower 35 per cent. larger, and withal the result of the higher vacuum and greater expenditure is a net plant gain of only 0.61 per cent.

In some cases the problem is not to increase the economies as much as to reduce the first cost. The cost of a cooling tower and condensing plant is often prohibitive, particularly in the case of low-pressure installations. Here also the cost of the turbine is greater, since the size is the same as the low-pressure portion of a high-pressure turbine of twice the capacity. Where these conditions prevail the two-stage condenser, turbine and tower are designed for lower first cost rather than higher economy.

If the turbine is operated at vacuums of 25½ and 27½ with a two-stage condenser, the water is heated up to 120 deg. and delivered to the tower much warmer, so that a large saving is made in the size of the tower and the fan power. The capacity and efficiency of a cooling tower depend largely on the temperature of the water to be cooled, as this in turn determines the temperature to which the air is heated and its capacity for taking up water vapor, and in a natural draft tower, the intensity of the draft.[2]

[2]This also applies to spray ponds, as shown by tests at Birmingham, Ala., reported by the Spray Engineering Co. The water was cooled from 112½ to 83.3 deg. in the first spraying, and from 88.3 to 78.8 deg. in the second spraying. The cooling in the first spraying with the warm water was almost three times as great as in the second.

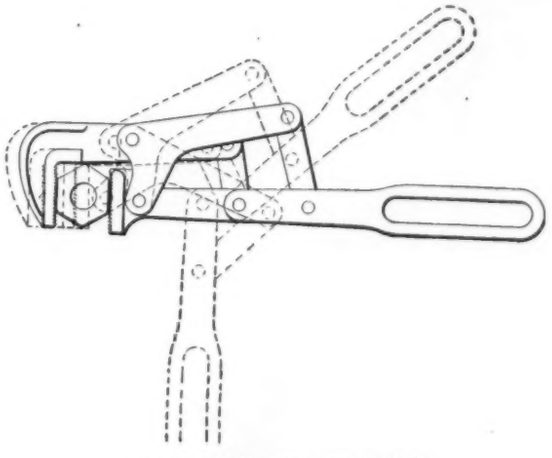
At the same time, with water heated in the condenser to 120 instead of 110 deg., less water is needed and less power is lost in pumping. Another saving results from the fact that the condensate is heated to a higher temperature. The net overall economy is thus increased slightly compared to operation at 27 in. on all the steam. See Case 3.

Compared to the straight 27-in. vacuum arrangement there is a saving of $2000 to $3000 in the cost of cooling tower and condenser, by operating at 25½ and 27½ in. in a two-stage condenser, using less condenser surface, a smaller circulating pump and piping, a smaller air pump, a cooling tower of only two-thirds the size and a turbine of one-third smaller final wheel area rated up to 1000 kw.

The fan power saving would not be included in plants using natural-draft towers, but even without a saving in fan power the table shows that the overall economy with 25½- and 27½-in. vacuums is still equal to that with 27 in., while at the same time the saving on the cost of a natural-draft tower would be greater. The tower would cost more, to start with, so that the saving would be greater and also the percentage reduction would be more than 33 per cent., because with warmer water and warmer exit air there would be a better natural draft and larger tower capacity. The total saving in cost in the tower would therefore approximate $2000, making the total saving in excess of $3000.

Hayward Nut and Pipe Wrench

The self-adjusting wrench shown herewith for bolts, nuts and pipe is made by the Hayward Wrench Co., St. Louis, Mo. The linkage used gives a nearly parallel motion to the movable jaw. The quick jaw movement makes it almost a ratchet in operation. The harder it is pulled, the tighter it holds. It is easy to operate on pipe or nuts, is simple and handy and is made of drop-forged steel.



SELF-ADJUSTING WRENCH

The Composition of Tobin Bronze is usually about as follows:

	Per cent.
Copper	59.00
Zinc	38.40
Tin	2.16
Iron	0.11
Lead	0.31

As cast metal it has a tensile strength of about 65,000 lb. per sq.in. of sectional area; when rolled, about 79,000 lb.; and when drawn down by cold rolling a tensile strength of over 100,000 lb. per sq.in.

Warren State Hospital Plant--II

By Warren O. Rogers

SYNOPSIS—Hot water for domestic service is heated by exhaust steam and held in storage tanks at 130 deg. F. Provision has been made for using live steam for heating the water when the demand is greater than the exhaust steam can supply at the required temperature. Steam at 150 lb. pressure is piped from the boiler house to the point of distribution, 600 ft. distant, through either of two 5-in. steam lines at a velocity of 16,000 ft. per min. and at a terminal pressure of 5 lb. or less. The result of the boiler tests showed that a full half-pound of water was evaporated per pound of combustible burned above the 11 lb. required. This earned the contractor a bonus of $300.

At the south end of the boiler room, at the Warren State Hospital, 12 ft. 3 in. below the floor level is a pit, in which two vertical storage tanks for domestic hot-water service,

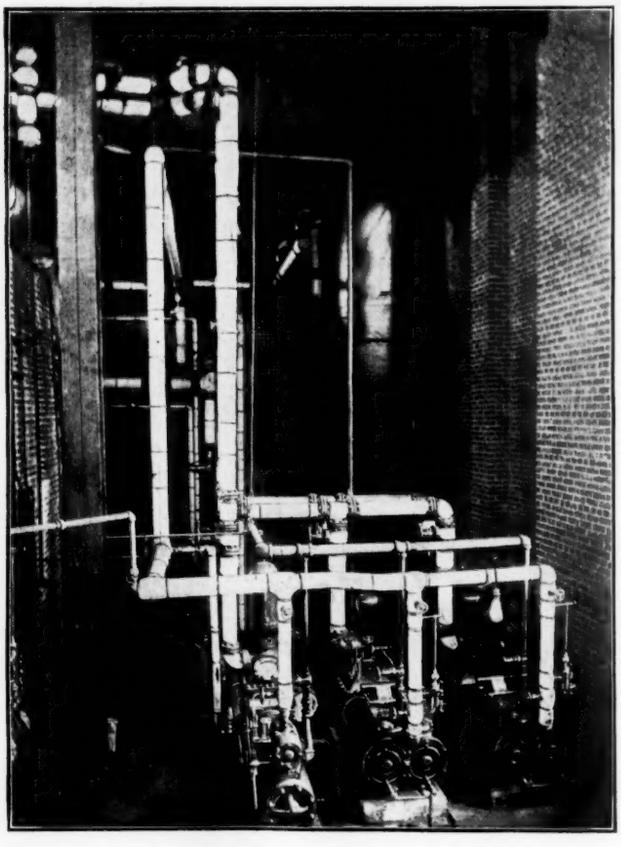


FIG. 8. BOILER-FEED PUMPS AND CONNECTIONS
Showing duplicate feed lines and hot-water storage tanks in the background

Fig. 8, are installed. The shells are 78 in. diameter and 32 ft. high. At the same end of the building, but on the engine-room floor level, are two steam, hot-water closed heaters, each having a capacity to heat 5,000 gal. of water per hr. from 50 to 150 deg. F. with steam at atmospheric pressure. Below these heaters, in the storage-tank pit, is a 3-hp. motor directly connected to a 2-in. volute pump. It has a capacity to handle 200 gal. of water per min. against a frictional head of 5 ft. and provides a forced circulation between the heaters and the storage tanks. On the floor-level alongside of the heaters is a second motor-driven volute pump of practically the same size as the other, with a capacity to handle 25 gal. of water per min. against a frictional head of 90 ft. It is used for maintaining constant circulation through the 5-

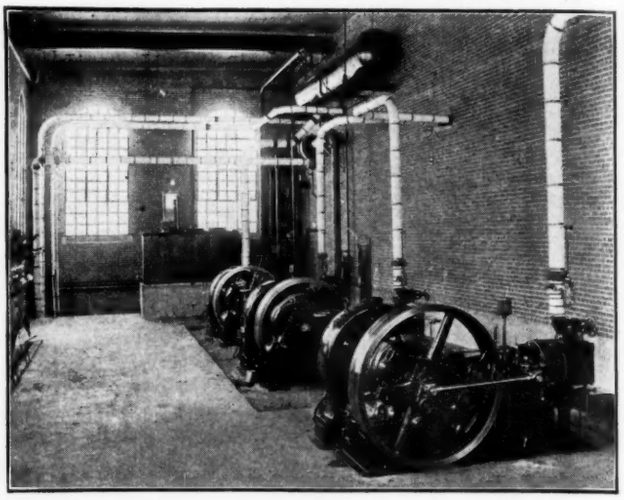


FIG. 9. VIEW OF ENGINE ROOM
Showing overhead return tank and feed-water meter at end of engine room; switchboard on the left, outgoing duplicate 5-in. steam lines and 5-in. hot-water lines entering tunnel.

in. domestic hot-water supply line running from the vertical storage tanks to the center of distribution at the old plant.

Automatic Temperature Control

The temperature of hot water in the storage tanks is automatically controlled. Under normal conditions the consumption of water is greatest when the engine load is lightest, and the supply of exhaust steam for passing through the closed heaters is not sufficient to heat the water to the required temperature when the hot-water consumption is high. In order to maintain the required water temperature a thermostat is arranged to control a regulating valve on the main exhaust line. There is also a thermostat on each storage tank placed about ten feet from the top, one being set a little ahead of the other. In case there is insufficient exhaust steam the thermostat that is set to operate at the lowest temperature operates a 2½-in. regulating valve in a live-steam line and steam is admitted to the tank until the temperature reaches 130 deg. If the water consumption is excessive, causing a large quantity of cold water to enter the tank and reduce the temperature, the other thermostat operates a second 2½-in. regulating valve on the second tank. When the consumption becomes normal and the temperature rises to 130 deg., the thermostats cut off the live-steam supply and the temperature is maintained through the exhaust-steam regulating valve.

The storage tanks have a 24-hr. water supply in case of emergency, and not only supply the institution with hot water, but also the boiler-feed pumps, the makeup and return water coming to the system through the feed-water

meter. The large storage supply is desirable, as all of the exhaust steam from the steam units at peak load can be absorbed instead of escaping to the atmosphere. This works to advantage, because when the peak lighting load is on the water consumption is least, and it is heaviest when the exhaust-steam supply is least.

At the south end of the engine room is a pump pit, Fig. 10, in which is a 10x16x18-in. simplex vacuum pump un-

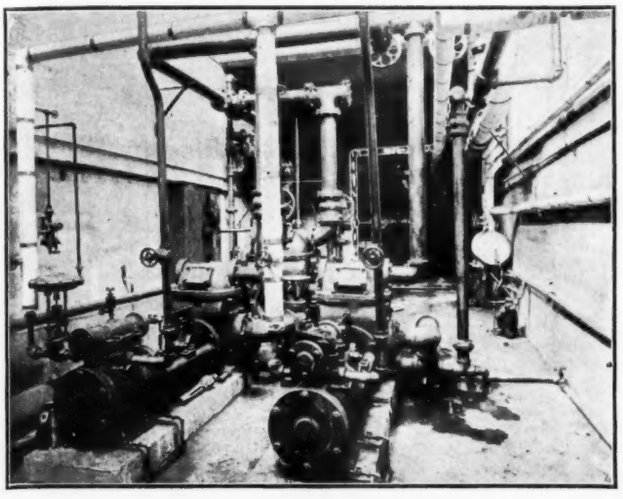


FIG. 10. VIEW IN PUMP PIT

Showing vacuum-return pumps and receiver for connection of high-pressure drips and power-house piping on the right.

der vacuum control. It handles the returns from the heating system from the group of buildings and also the return condensation from the domestic hot-water heaters and discharges into the 42-in. diameter by 18-ft. long return tank already mentioned. The tank is provided with a float valve for controlling the supply of makeup water. There are also two vacuum pumps, brought

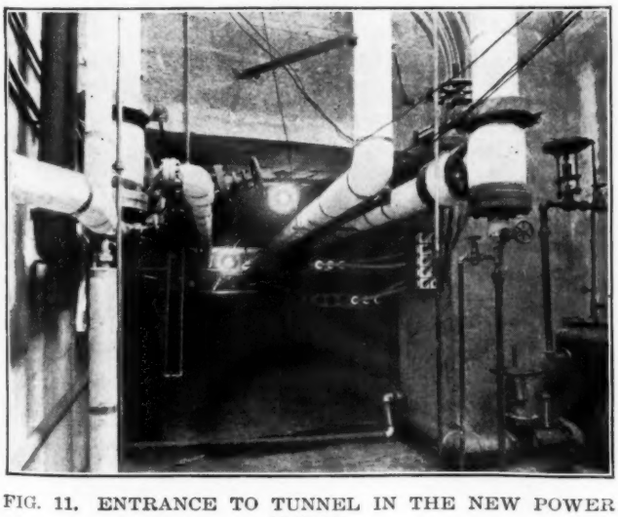


FIG. 11. ENTRANCE TO TUNNEL IN THE NEW POWER HOUSE

from the old plant to act as reserve units, the two having a capacity equal to the simplex pump. The pipe connections are so arranged that one of the old vacuum pumps can be used for emptying the domestic hot-water storage tanks when they require cleaning.

The concrete tunnel between the new and old power houses is 8 ft. wide, 7 ft. high and 590 ft. 6 in. long. In it are run the two 5-in. and the 3-in. steam and the 5-in. hot-water domestic-supply lines, the return-pipe line and the electric-feeder lines connecting the switchboard in the new plant with the distributing board in the old station. Figs. 11 and 12 show the entrances to the tunnel in the new and old power houses, respectively.

At the right of Fig. 11 is the condenser to which is connected the 5-in. return line. The regulating valve is set at 9 in. vacuum and those on the vacuum pumps at 10 in. If the pump loses its vacuum the regulating valve on the cold-water pipe connecting to the condenser admits cold water to decrease the temperature. The drips from the heater are trapped and the condensate is discharged to the condenser, from which it is taken by the vacuum pumps with the return from the heating system.

Arrangement of Main Piping System

All high-pressure steam piping 4 in. and larger in the new power house and all connections to the old work already in place are made up with Van Stone joints and

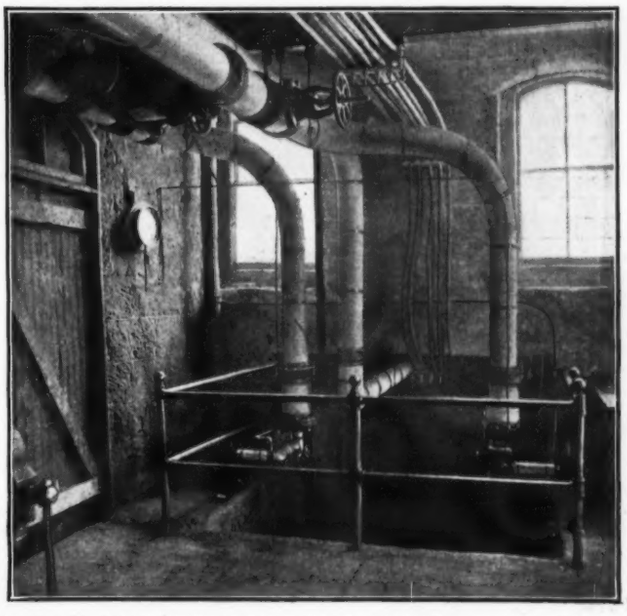


FIG. 12. ENTRANCE TO TUNNEL IN OLD POWER HOUSE

extra-heavy flanges. All high-pressure steam piping below 4-in. is put together with extra-heavy cast-iron fittings; 2-in. pipes and smaller are of wrought iron.

A 10-in. welded-steel header, anchored in the center to prevent longitudinal movement and near each end to prevent lateral movement, runs back of the boilers, and 6-in. vertical lines branch from it to a point convenient for coupling to the 5-in. connections to the boilers. From the header three 4-in. lines run to the three engines. A 3-in. pipe branches from the south end of the 10-in. header and supplies steam to the three boiler-feed pumps. The vacuum pumps in the pump pit are supplied through a 2½-in. pipe.

From the south end of the header a 6-in. line is run to the tunnel side of the engine room, Fig. 9, where it divides into two 5-in. lines which drop to the pump pit and extend through the tunnel to the old power plant. Each is anchored at the ends and at the turn of the tunnel. These lines are each fitted with six expansion joints spaced at equal distances in the tunnel.

The two 5-in. steam pipes are smaller than are usually employed for steam heating under similar conditions.

The steam velocity is about 16,000 ft. per min., or 266 ft. per sec., with an initial pressure of 150 lb. at the boilers, and reduces to a terminal pressure of 5 lb. or less. As this reduction of pressure takes place in the 600 ft. of steam lines, no reducing valves are used and the small size of the feed lines materially reduces the initial cost of installation. The duplicate steam lines insure a continuous supply for heating the buildings. In case of the main header back of Nos. 1 and 2 boilers becoming disabled, thus shutting off the steam supply to the 6-in. line that supplies the two 5-in. steam lines, steam can be supplied through the 5-in. auxiliary pipe that connects

DAILY LOG OF POWER PLANT

STATE HOSPITAL FOR THE INSANE, WARREN, PA.

DATE _____ 191

	Time	BOILER PLANT										ELECTRIC PLANT									HEATING			HOT WATER			
		BOILERS		Coal Lbs.	Ash Lbs.	FEED WATER			Steam Pres.	Draft		Units		OUTPUT							Returns		Exhaust	Meter		Temp.	
		Started	Stopped			Meter Index	Lbs.	Temp.		Breeching	Over Fire	Started	Stopped	Volts	AMPERES No. 1	No. 2	No. 3	K. W. H. No. 1	No. 2	No. 3	Temp.	Vacuum	Back Pres.	Index	Lbs.	Outlet	Inlet
	12 M																										
	1 a.m.																										
	2																										
	3																										
	4																										
	5																										
	6																										
	7																										
	8																										
	9																										
	10																										
	11																										
	12 N																										
	1																										
	2																										
	3																										
	4																										
	5																										
	6																										
	7																										
	8																										
	9																										
	10																										
	11																										

SUMMARY

	Total Coal Lbs.	Cyl. Oil Qts.	Outside Temp.	REMARKS
	Total Ash Lbs.	Engine Oil Qts.	12.00 M	
	Total Water Evap. Lbs.	Supplies	6.00 A. M.	
	Av. Temp. Feed		12.00 Noon	
	Factor of Evap.		6.00 P. M.	
	Equiv. Evap. from and at 212°, Lbs.	Repairs		Engineer on Duty
	Total Boiler H. P. Hrs.			
	Total K. W. Hrs.			

FIG. 13. DETAILS OF DAILY LOG SHEET OF THE POWER PLANT

WEEKLY SUMMARY OF LOG OF POWER PLANT

STATE HOSPITAL FOR THE INSANE, WARREN, PA.

WEEK ENDING 12 M. SATURDAY _____ 191

	Day	BOILER PLANT											ELECTRIC PLANT							HEATING			HOT WATER		
		Coal Lbs.	ASH		FEED WATER		Av. Steam Pres.	Factor of Evap.	Equiv. Evap. from and at 212°, Lbs.	Evap. per lb. Coal from and at 212°	Boiler H. P. Hours	Coal per B.H.P.H. Lbs.	Total K. W. H.	Max. Amps	Min. Amps	Av. K. W.	Cyl. Oil Qu'ts	Eng. Oil Qu'ts	Returns		Exhaust	Lbs.	Av. Temp. Outlet	Av. Temp. Inlet
			Lbs.	%	Lbs.	Av. Temp.													Av. Temp.	Av Vacuum	Av Back Pres.			
	Sunday																							
	Monday																							
	Tuesday																							
	Wedn'sday																							
	Thursday																							
	Friday																							
	Saturday																							
	Totals																							

REMARKS

FIG. 14. WEEKLY LOG SHEET USED AT THE POWER PLANT

with the main header between Nos. 3 and 4 boilers. This arrangement of piping is flexible, insures a constant steam supply and was adopted after considerable investigation on the part of the consulting engineer.

Result of Boiler Test

Upon the completion and putting in operation of the new power plant a boiler test was made to ascertain the economy and capacity. Because No. 1 boiler could be isolated more readily than either of the others, it was selected for the test, which was run in accordance with the revised code of the American Society of Mechanical Engineers, using the alternate method of starting and stopping the trial.

During the evaporation tests it was required that the boiler evaporate water at the rate of 3.6 lb. per hour from and at 212 deg. F. per sq.ft. of heating surface, or a total of 4531 × 3.6 = 16,312 lb., and when operating at this rate of load the performance of the boiler was to be at the rate of 11 lb. of water evaporated from and at 212 deg. F. per pound of combustible burned.

During the test the average rate of evaporation was 11.725 lb. of water per pound of combustible, an excess of 0.725 lb. As the contract stipulated that a bonus of $100 per boiler equipped with stokers would be given for each full ½ lb. of water evaporated over and above 11 lb., a bonus of $300 was earned, as shown by the results of the test.

The requirements of the flue gases during full-load tests were that they should contain not less than 12 per cent. CO_2 and not more than 0.2 per cent. CO. The result of the test showed an average of 12.79 per cent. CO_2 and 0.07 per cent. CO.

It was required, with the stoker operating with the rated load on the boiler, that not more than 3 per cent. of the coal dry should show as combustible in the ash. An analysis of the ash after the test showed that 2.135 per cent. of the coal dry showed as combustible.

The overload test required that the boiler evaporate 5.4 lb. of water per hour from and at 212 deg. F. per sq.ft. of heating surface. The test showed an equivalent evaporation of 5.858 lb.

Fig. 15 is a combination chart of the coal and water consumption per hour, steam pressure, temperature of flue gases, draft inches in the combustion chamber and boiler uptake, temperature of feed water, per cent. CO_2 and O, also total coal and water used during the test.

Details of the evaporation test and overload boiler test are given in the following:

RESULTS OF EVAPORATION TEST

Made by J. E. Woodwell, of No. 1 Geary water-tube boiler and Detroit automatic mechanical stoker at the State Hospital for the Insane, Warren, Penn., to determine the evaporative performance, efficiency, capacity, etc.

Kind of fuel—Crushed three-quarter bituminous coal mined at Sterling Run, Cameron County, Penn.

Kind of furnace—Detroit Stoker Co.

State of the weather—Fair.

Method of starting and stopping the test (standard or alternate)—Alternate.

Date of trial—1 p.m., Nov. 19, to 1 p.m., Nov. 20, 1914.

Duration of trial—24 hr.

Dimensions and Proportions	
Grate surface, width 11 ft. 2 in., length 7 ft. 2 in., area	80 sq.ft.
Grate surface (projected), width 9 ft. 8 in., length 7 ft. 2 in., area	69.3 sq.ft.
Height of furnace from lower end of grates to top of arch	76 in.
Approximate width of air spaces in grate	⅝ in.
Proportion of air space to whole grate surface	32 per cent.
Water-heating surface	4531 sq.ft.
Ratio of water-heating surface to grate surface	56.64 to 1
Ratio of minimum draft area to grate surface	1 to 3.56

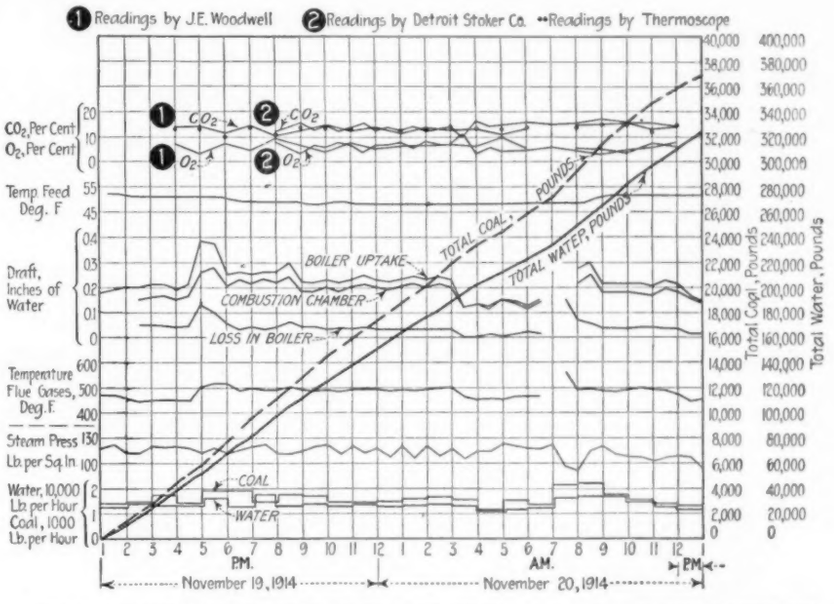


FIG. 15. COMBINATION CHART OF THE COAL AND WATER CONSUMPTION, STEAM PRESSURES, TEMPERATURE OF FLUE GASES AND FEED WATER AND THE DRAFT

Average Pressures	
Steam pressure by gage	118.5 lb. per sq.in.
Force of draft between damper and boiler	0.2281 in. of water
Force of draft in furnace	0.1900 in. of water
Force of draft or blast in ashpit	0.0000 in. of water

Average Temperatures	
External air	32.2 deg.
Fireroom	67.3 deg.
Steam	348.82 deg.
Feed water entering boiler	49.3 deg.
Escaping gases from boiler	479.3 deg.

Fuel	
Size and condition	Crushed three-quarter coal
Weight of coal as fired	36,746 lb.
Percentage of moisture in coal	4.02 per cent.
Total weight of dry coal consumed	35,269 lb.
Total ash and refuse	2,633 lb.
Total combustible consumed	32,636 lb.
Percentage of ash and refuse in dry coal	7.465 per cent.

Proximate Analysis of Coal	
Fixed carbon, Volatile matter	88.94 per cent.
Moisture	4.02 per cent.
Ash	7.04 per cent.
	100.00 per cent.
Sulphur, separately determined	0.80 per cent.

Analysis of Ash and Refuse	
Carbon	28.61 per cent.
Earthy matter	71.39 per cent.

Fuel per Hour	
Dry coal consumed per hour	1469.5 lb.
Combustible consumed per hour	1359.8 lb.
Dry coal per sq.ft. of grate surface per hr.	18.37 lb.
Combustible per sq.ft. of water-heating surface per hour	0.30 lb.

Calorific Value of Fuel	
Calorific value by oxygen calorimeter, per lb. of dry coal	13,778 B.t.u.
Calorific value by oxygen calorimeter, per lb. of combustible	14,869 B.t.u.

Quality of Steam	
Percentage of moisture in steam	0.58 per cent.
Percentage of leakage	1.80 per cent.
Quality of steam (dry steam = unity)	99.42 per cent.

Water	
Total weight of water fed to boiler	321,960 lb.
Water apparently evaporated corrected for leakage	317,678 lb.
Water actually evaporated, corrected for quality of steam	316,312 lb.
Factor of evaporation	1.2097
Equivalent water evaporated into dry steam from and at 212 deg.	382,643 lb.
Water per Hour	
Water apparently evaporated per hour corrected for leakage	13,237 lb.
Water evaporated per hour corrected for quality of steam	13,180 lb.
Equivalent evaporation per hour from and at 212 deg.	15,943 lb.
Equivalent evaporation per hour from and at 212 deg. per sq.ft. of water-heating surface	3.519 lb.
Horsepower	
Horsepower developed (34½ lb. of water evaporated per hour into dry steam from and at 212 deg., equals 1 hp.)	462.12
Builders' rated horsepower	453.10
Percentage of builders' rated horsepower developed	101.99 per cent.
Economic Results	
Water apparently evaporated under actual conditions per lb. of coal as fired.	8.762 lb.
Equivalent evaporation from and at 212 deg. per lb. of coal as fired	10.413 lb.
Equivalent evaporation from and at 212 deg. per lb. of dry coal	10.849 lb.
Equivalent evaporation from and at 212 deg. per lb. of combustible	11.725 lb.
Efficiency	
Efficiency of the boiler	76.52 per cent.
Efficiency of boiler, including the grate	76.41 per cent.
Analysis of Dry Gases	
Carbon dioxide (CO_2)	12.79 per cent.
Oxygen (O)	5.73 per cent.
Carbon monoxide (CO)	0.07 per cent.
Nitrogen (by difference) (N)	81.41 per cent.

RESULTS OF OVERLOAD TEST

Report and data of overload boiler test, State Hospital for the Insane, Warren, Penn., made by J. E. Woodwell, of No. 1 Geary water-tube boiler and Detroit automatic mechanical stoker, at the State Hospital for the Insane, Warren, Penn., to develop 50 per cent. overload.

Kind of fuel—Slack coal mined at Braznell mine in Youghiogheny gas slack district near Pittsburgh, Penn.

Kind of furnace—Detroit automatic mechanical stoker.

State of weather—Fair.

Method of starting and stopping the test (standard or alternate)—Alternate.

Date of trial—2:30 p.m. to 4:30 p.m., Nov. 20, 1914.

Duration of trial—2 hr.

Average Pressures	
Steam pressure by gage	117.1 lb. per sq.in.
Force of draft between damper and boiler	0.721 in. of water
Force of draft in furnace	0.697 in. of water
Average Temperatures	
External air	26.4 deg.
Fireroom	55.6 deg.
Steam	348.07 deg.
Feed water entering boiler	51.0 deg.
Escaping gases from boiler	634.2 deg.
Fuel	
Size and condition	Slack coal
Proximate Analysis of Coal	
Fixed carbon	51.72 per cent.
Volatile matter	30.42 per cent.
Moisture	3.09 per cent.
Ash	14.77 per cent.
	100.00 per cent.
Sulphur, separately determined	1.61 per cent.
Calorific Value of Fuel	
Calorific value of oxygen calorimeter, per lb. of dry coal	12,720 B.t.u.
Calorific value by oxygen calorimeter, per lb. of combustible	15,007 B.t.u.
Quality of Steam	
Percentage of moisture in steam	0.55 per cent.
Percentage of leakage	1.08 per cent.
Quality of steam (dry steam = unity)	99.45 per cent.
Water	
Total weight of water fed to boiler	44,480 lb.
Water apparently evaporated, corrected for leakage	44,124 lb.
Water actually evaporated, corrected for quality of steam	43,943 lb.
Factor of evaporation	1.208 lb.
Equivalent water evaporated into dry steam from and at 212 deg.	53,083 lb.
Water apparently evaporated per hour, corrected for leakage	22,062 lb.
Water evaporated per hour, corrected for quality of steam	21,972 lb.
Equivalent evaporation per hour from and at 212 deg.	26,542 lb.
Equivalent evaporation per hour from and at 212 deg. sq.ft. of water-heating surface	5.858 lb.
Horsepower	
Horsepower developed (34½ lb. of water evaporated per hour into dry steam from and at 212 deg., equals 1 hp.)	769.3
Builders' rated horsepower	453.1
Percentage of builders' rated horsepower developed	169.8 per cent.

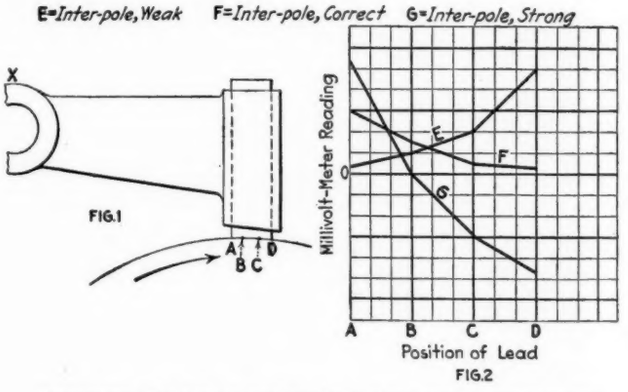
In the foregoing tests the contractors were represented by B. B. Freer, general sales manager for the Oil City Boiler Works; Edwin Fitts, supervising engineer for the Detroit Stoker Co.; and the consulting engineer by J. E. Woodwell and C. A. Bulkeley. Upon the completion of the plant Mr. Woodwell was engaged as supervising engineer for the institution.

Neutral on Interpole Machines

By E. T. Keenan

The function of the interpole in an interpole motor is to produce in the armature coil under commutation an electromotive force of the right value and sign to reverse the current in that coil while it is yet under the brush.

Self-induction has the effect of opposing any change in direction due to a change in the strength of the current. Hence, ordinarily the brush would shift to a point where an electromotive force is generated in the coil from the flux that will be equal to the electromotive force of self induction and no sparking will occur. This point will change, however, with every change of load or strength



POSITION OF LEAD ON BRUSH AND CORRESPONDING VOLTAGE READINGS

of current through the armature or fields as any change in strength will change the value of the electromotive force of self-induction. Therefore, to obviate the necessity of constantly shifting the brushes, modern practice is to change the strength of the flux by means of additional poles (interpoles) in series with the armature. It is apparent that the strength of this pole is very important, and the right strength may be found as follows:

Connect one lead from a millivoltmeter to the metal of the brush holder as at *X*, Fig. 1, and the other lead to the heel of the brush, as at *A*, and note the reading. Now change this lead to a point *B;* then to *C* and *D*, the toe of the brush. Note the voltage at each position and plot these as a curve on cross-section paper and compare with Fig. 2.

The polepieces of this type of machine are placed with thin shims, and to vary the strength of the poles add to or take from the number of shims. Great care must be exercised in doing this. Care is taken in the factory to get them right, and only occasionally does one pass that

is not correct. The brush-holder yoke of this type of machine is always doweled in place, and it is not intended that the brushes should be moved; but as the machines are tested in groups and not separately, brushes have been found that were not set correctly.

The neutral point can be readily determined by the use of a millivoltmeter as follows: Take two hard lead pencils and sharpen them at each end. To one end of each fasten one lead of the millivoltmeter, and bind the pencils side by side close enough together so that the distance between the centers of the leads is the same as the distance between the centers of adjacent commutator segments. With the pencil points resting on two adjacent segments send a current at about one-fourth or one-third of full voltage through the fields and break the circuit quickly. This will cause an induced electromotive force in the armature coils, and the sign of this electromotive force will depend on which side of the neutral point the pencils have made contact on the commutator.

If the coil whose leads are connected to the segments on which the pencils rest is in the neutral position, no deflection will show. If the millivoltmeter has its zero in the middle and reads each way, it will be necessary only to move the pencils one bar every time the circuit is broken. When a point is reached where the needle stays at zero, make a convenient mark and place the brushes there. Remember that the brushes must be raised off the commutator when the fields are energized, and do not use full voltage, as the high-induced electromotive force caused by breaking the circuit might puncture the insulation.

Tracy Steam Purifier

Every engineer desires to obtain dry clean steam, and any device designed to produce it is of interest. One such apparatus is shown in Fig. 3, the Tracy steam purifier, manufactured by the Tracy Engineering Co., 461 Market St., San Francisco, Calif.

The apparatus consists of a shell in which is suspended a vertical-gutter baffle-plate, Fig. 2. The steam enters

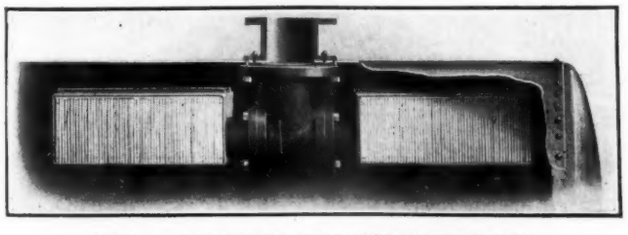


FIG. 1. PURIFIER FOR STEAM BOILERS

the space between the gutters in the first row, and is divided into thin vertical streams, which flow toward the open side of the gutters in the second row, where each ribbon of steam is divided into two thinner streams which make a right-angle turn in passing through the narrow vertical ports constituting the space between the two rows of gutters.

After passing these ports the steam goes through the remaining series of gutters to reach the central conduit leading to the boiler outlet. In this way the steam is divided into numerous thin streams so small that a particle of water has but a short distance to travel to get out of the steam into the protection of the gutters where there is no steam current. The aggregate area of the ports is several times that of the boiler nozzle, which gives a low velocity to the steam in passing through the purifier.

Purifiers for steam boilers, Fig. 1, are made to pass through a 10x15-in. manhole, and in a length that can be handled in restricted spaces in the boiler. For this type a cast-iron nozzle tee or elbow is provided for at-

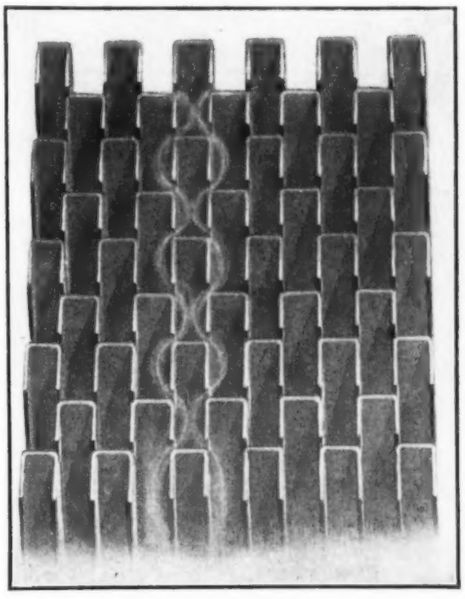


FIG. 2. VRETICAL-GUTTER BAFFLE-PLATE

FIG. 3. INTERIOR OF THE PURIFIER

taching to the main steam outlet and the purifier elements are bolted to the tee, the outer element, when necessary, being supported by a bracket. After the purifier is installed it requires no attention, as the water drains to the boiler.

Tube Surface in Condensers—Surface condensers are allowed from 1½ to 2½ sq.ft. of tube surface per indicated horsepower or, better, 1 sq.ft. tube surface for 8 to 12 lb. of steam to be condensed per hour.

Operation at Delray Station*

By Norman G. Reinicker

SYNOPSIS—Predicting the load for a given day, determining the number of boilers and turbo-generators to operate and how the firemen maintain the service.

Due to the character of the business served by the company, the load on the system can be predicted closely for any day of the week or season of the year. A predicted load curve, drawn by the routine-test department, is given regularly to the operating engineer. From predicted load curve (see chart shown in Fig. 1) the operating engineer determines the minimum, average and maximum amounts of steam the boiler room will be required to furnish. Knowing the amount of steam required, he decides on the number of boiler units necessary to operate to generate this steam and yet keep the boilers running for the greatest part of the time at their most efficient point. In counting the number of boiler units necessary, one large boiler is considered the equivalent of five small ones, since the large boilers have practically five times the heating surface of the small ones, and the ratio of grate surface to heating surface is practically the same in both sizes.

Repeated tests and actual operating results have shown that any of the boilers in the plant can be operated from 70 per cent. to over 200 per cent. of rating efficiency. Advantage is taken of this in the operation of the plant.

The actual efficiency curves of both the large and the small boiler units under test conditions show a maximum efficiency of 80 per cent. at less than rating (rating being considered the evaporation of 3.45 lb. of water per square foot of heating surface per hour from and at 212 deg. F.). In actual operation, however, the best efficiency seems to be attained at from 125 to 130 per cent. of rating. This is true, because with this load the fires are more easily controlled and therefore more likely to be burning under the most efficient conditions the greater part of the time. During the "day load" just enough boilers are operated to maintain an average rating of or nearly 130 per cent. During the "peak load" there is enough boiler capacity already in the system, so that the "peak" can be carried by merely raising the rate of steaming of the boilers that are already on the system. During the "day load" no banked fires are carried. At night as many boilers as possible are kept steaming, running them at as low a rating as is practical. The rest of the boilers must be banked.

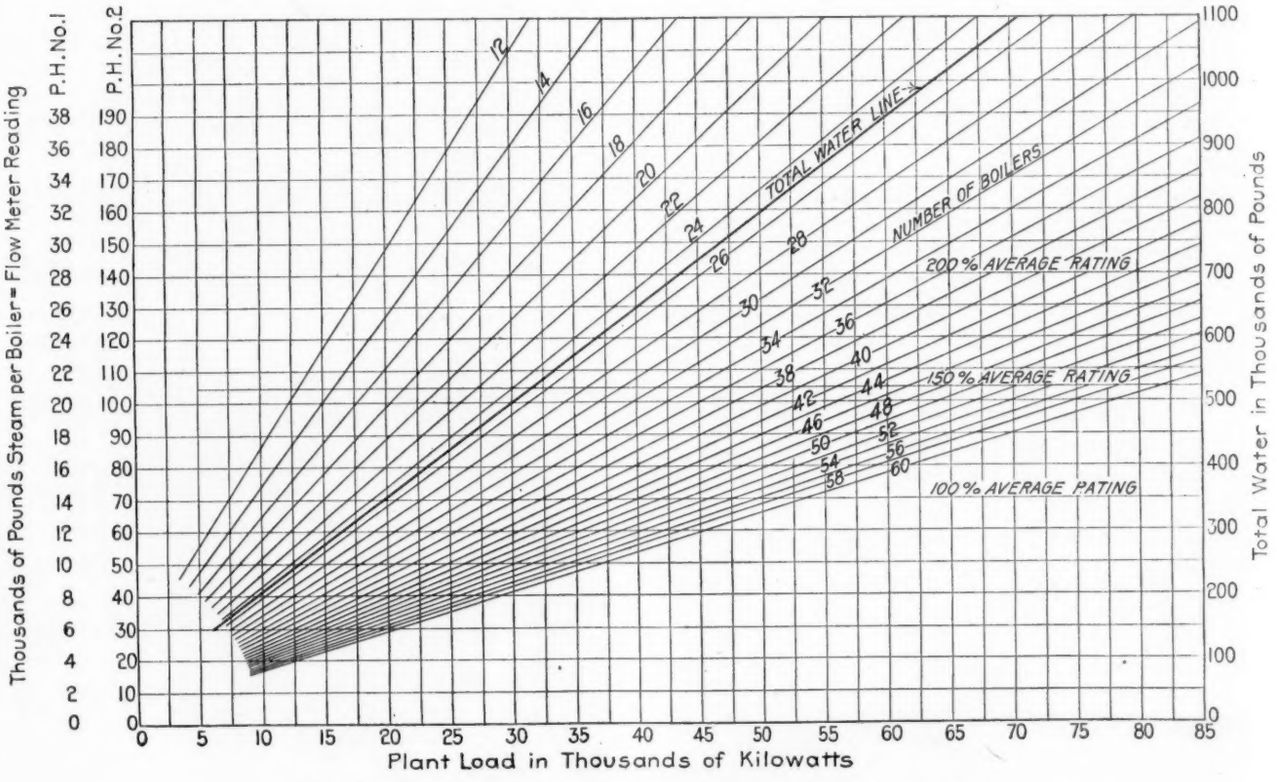

Thousands of Pounds Steam per Boiler = Flow Meter Reading
P.H. No.1
P.H. No.2
Total Water in Thousands of Pounds
Plant Load in Thousands of Kilowatts
TOTAL WATER LINE
NUMBER OF BOILERS
200% AVERAGE RATING
150% AVERAGE RATING
100% AVERAGE RATING

FIG. 1. POUNDS OF STEAM PER BOILER FOR ANY PLANT LOAD

Evening Peak Load Curve

Figure 2 (the load curve for Dec. 9, 1914) shows the evening peak where the power and lighting loads overlap. On the same sheet there is drawn a chart showing the number of boilers steaming and their rating for the different loads. The same number of boilers is maintained on the system at all times of the day, the fluctuation in the load on the plant being cared for by changing the rate of steaming. Fires are banked only when the rate of steaming falls below 80 per cent.

*See also, by the same author, "Delray Power Plant," Aug. 31, p. 286, and "Organization at Delray," Sept. 7, p. 343.

This chart shows fifty boiler units (one large boiler equals five units) with fires in their furnaces during the entire twenty-four hours of the day, the boilers being operated from a minimum of 81 per cent. of rating to a maximum of 179 per cent. during the peak load. It is also apparent from this figure that at any time of the day the whole plant is in such a condition that one of the large boilers (rated at 2365 hp.) can either as a result of an accident or by prearrangement be dropped off the system, and the rest of the boilers will still have enough capacity to carry the peak load without undue forcing. The data for the load curve and the boiler chart were taken from the power-plant operating log sheet for the date previously given.

The flexibility of the Taylor stoker is shown by the ease with which these wide variations in load are carried. The drop in the load at noon and the evening peak are

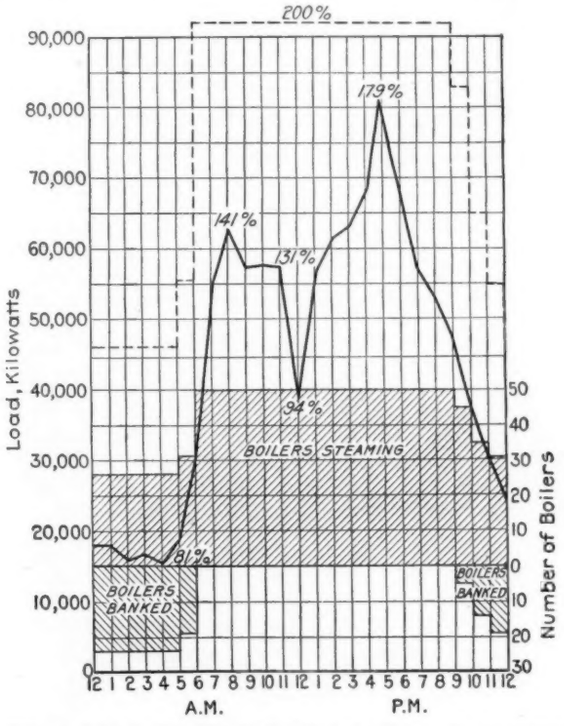


FIG. 2. LOAD CURVE FOR DEC. 9, 1914, AND BOILER CHART

carried with the same degree of ease, and the transition from one to the other involves no more increased activity in the boiler room than that actually required for normal operation.

The operating engineer is supplied also with a schedule giving the number of turbo-generators to run and their loading for any plant load. Several elements govern the making of this schedule. The routine-test engineering department determines the water rates of all the main units. In this connection a predetermined policy is observed regarding the factor of safety in the capacity of the machines in the system. This policy dictates that it should be possible in case of an emergency to trip out the largest machine in the system, and the remaining machines will still have sufficient capacity to carry the load. In loading the machines the operating engineer is guided by the schedule but in deciding on the best loading point he frequently deviates from schedule, as dictated by test results or by some condition arising at the time.

The boiler-room foremen advise the firemen of their allotted share of the load to be carried for the day, translated into pounds of steam as measured by the flow meters on the boiler outlets, the capacities being varied slightly from time to time as found necessary to maintain a constant steam pressure at the turbine throttles. The foreman is able to watch this steam pressure closely on a large pilot pressure gage mounted on a board at the end of the firing aisle, upon which is also placed an electrically operated clock and a manually controlled load indicator operated from the switchboard gallery. These instruments are shown on the board at the end of the firing aisles in Figs. 4 and 5 of the first article of the series, page 288, Aug. 31, 1915.

A board similar to this is installed at the end of each of the four firing aisles in the plant. The pilot gage is 30 in. in diameter, and on the dial are unnumbered divisions, 2 in. wide, each representing a 1-lb. change in pressure. A change in pressure as small as ¼-lb. is easily detected with this instrument, and the firemen are thus enabled to keep in intimate touch with the change of conditions in the steam piping system. The load indicator is set by the switchboard operator to read the total load at any time on all the machines in the plant. Being informed of the load on the plant from time to time, the fireman soon finds an explanation for any undue change in steam pressure which is causing him materially to alter the rate of steaming of his boilers from that determined by the operating engineer for the predicted load.

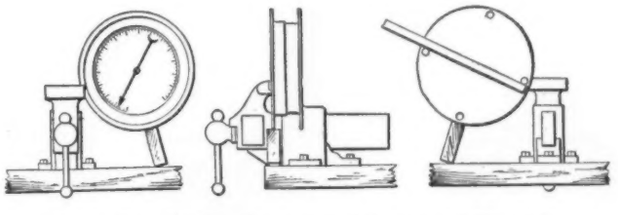
The general form of the load curve does not change for the different days of the week, so that the men in the plant know approximately what to expect at the different hours, and by watching the clock can keep their fires in condition to take care of an increase or decrease in the load, as it may be expected.

Inspecting Steam Gages

By F. W. Bentley, Jr.

Steam-gage dial glass casings often become corroded and difficult to unscrew from the body of the gage, partly because they are hard to get a good hold on or to apply any strong gripping device without doing damage.

The illustration shows how they can be removed with little trouble. One side of the cover is pinched between



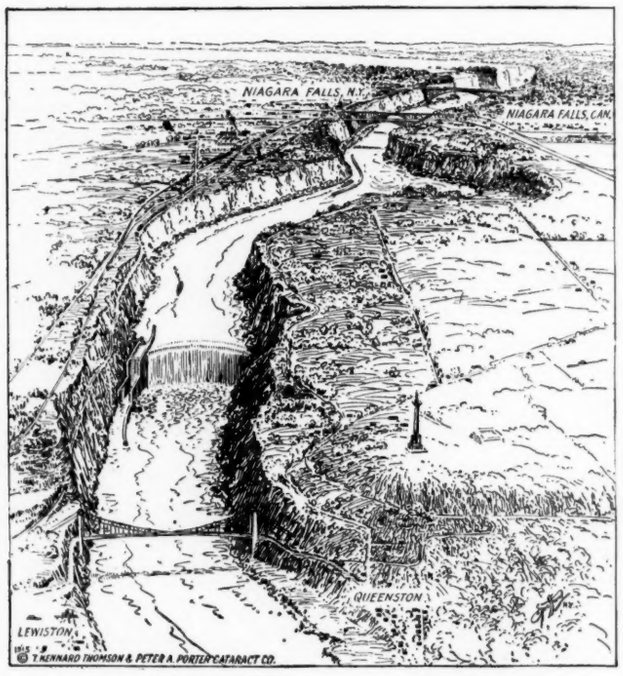
GAINING ACCESS TO INTERIOR OF GAGE DIAL

the projecting jaws of a vise and the lower portion rested on a block of wood so that the back or body of the gage is free to turn. With two bolts in the bracket holes of the body and a narrow strip of iron used as a lever almost any casing can be started, after which it can be readily taken off by hand. Getting it started is where the difficulty lies; but the most obstinate cover can be removed in this manner without danger of disfiguring it.

New Niagara Power Project

SYNOPSIS—A new Niagara power development is suggested in which a 90-ft. dam would be built several miles below the falls.

To the joint legislative committee on the Niagara situation assembled at Niagara during the first week in August, Peter A. Porter presented a plan to place a dam 90 ft. high across the Niagara River near Lewiston (about

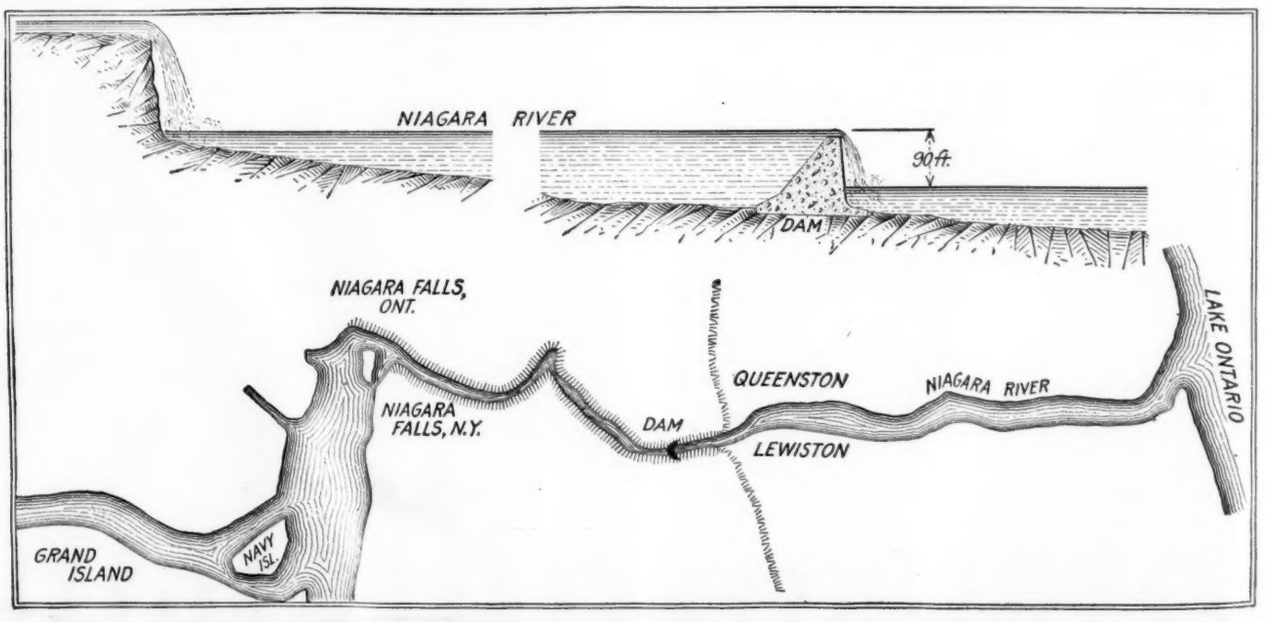


BIRD'S-EYE VIEW OF NIAGARA RIVER

7 mi. below the falls) at a point where the river is very narrow and the steep banks of natural rock about 300 ft. high afford the best opportunity for a dam of enormous strength. The plan which Mr. Porter presented is that of T. Kennard Thomson, a consulting civil and mechanical engineer of New York City.

It is estimated by the promoters that from 1 to 2 million horsepower will be available at this Niagara Falls, Jr., and the cost of the undertaking is estimated at 100 million dollars. It is suggested that the State of New York and the Province of Ontario undertake the project jointly or grant the right to a private corporation which would pay a fixed tax on the amount of power developed. At the rate of $5 per hp. per annum, the revenue for the State of New York would be nearly 5 million. By the use of the flow of this river for power a saving of 50 million tons of coal per year would be effected—a far-reaching economic consideration. The impounding of the water at this point to the depth of 90 ft. would not interfere in any way with the five great power plants now existing, which are developing about 500,000 hp.—nearly the maximum of water diversion permitted by law. Therefore any future development must be below the present falls. The advocates of the plan acknowledge that the grandeur and scenic beauty of the rapids and whirlpool would be destroyed, but contend that the creation of a new falls and the benefits derived by the use of the enormous power of the river will compensate fully for the loss. The new project is being pushed by the same man, Peter A. Porter, who in 1886 in the New York State Assembly introduced the bill which led to the construction of the present Niagara tunnel and power plant.

The Lake Survey Office of the United States War Department states that "the discharge of the Niagara River as determined at points above Lewiston is 210,000 cu.ft. per sec. when Lake Erie is at its mean stage, 572.55 ft. above mean sea level. The discharge increment per foot rise of Lake Erie is 22,100 sec.-ft. The monthly means have at times been 1.85 ft. below and 1.97 ft. above the mean stage and for a few hours even higher or lower than these." It appears then that the average possible with a 90-ft. fall and a 70-per cent. efficient waterwheel would be about 1¼ million horsepower.



ELEVATION AND COURSE OF NIAGARA RIVER, LOCATION OF PROPOSED DAM

Practical Use of Thermometers in Refrigerating Plants

By Peter Neff

SYNOPSIS—You would not run an electric generator without some means of indicating the voltage, at least. It would be just as inconsistent to run a refrigeration system without thermometers. Yet this is frequently done. The author suggests many things of practical interest about thermometers in this article.

In many plants the absence of thermometers is noticeable. They can usually be found in cold-storage rooms and ice tanks, but aside from this they have little general use, and yet here is a work that is altogether a matter of heat transference, and certainly, if this process is to be carried on with any degree of accuracy the temperatures in the different parts of the system must be known.

In refrigeration it is axiomatic that all the work possible be done by natural means before resorting to mechanical. This implies that everything from which heat is to be extracted should be acted upon by water or air or a combination of both before beginning to remove the heat by the use of a machine.

Consider the case of distilled water in an ice plant. This starts on its course boiling and is cooled by water; but how often is this only partly accomplished, and in how few plants is any record kept of the temperature to which the distilled water is cooled! If you ask the temperature of the water used for cooling purposes, in many plants you will be told that it is "about so and so," but they are not sure. The distilled water should be cooled to within at least 2 deg. F. of the water available for cooling purposes. How is this to be known unless there are thermometers so placed as to give these temperatures?

Preferable Thermometer Mountings

Here the writer wishes to digress for the purpose of saying something about thermometers to be used about the plant. If you can afford the fixed type they are convenient, provided they are so connected that in case one should break it can be removed from its well and another one substituted without disturbing the apparatus to which it is connected or in which it is inserted. For if the ordinary screwed thermometer is used and it should break, the time could not be taken to shut off the line to replace it. What gives equally as good results and at somewhat less cost is to insert thermometer wells at the points where temperatures are desired and then use chemical thermometers in these wells. These are inexpensive and accurate. Naturally, many get broken, and an extra supply should be on hand.

To return to the distilled water, knowing the temperature at which it leaves the cooler, it is quite essential that in its passage it does not have a chance to absorb heat. For this reason it is desirable to have a thermometer on the line leading from the storage tank to the can filler. In the event of precooling the water in the storage tank, this will then serve to indicate the amount of cooling done.

It is not advisable to attempt to get the temperature by holding the thermometer in the water-distributing trough or in a stream of waste water—there are too many opportunities for wide variation. Thermometer wells should be placed in the lines so that the water circulates freely about them.

It may reasonably be asked what all this amounts to. It is no uncommon thing to find the distilled water going to the storage tank 10 deg. warmer than the water used to cool it. In many plants it will be found to be much more than this.

Saving in 50-Ton Ice Plant

In a 50-ton ice plant, where 100,000 lb. of water are used per day, this 10 deg. represents about two tons of ice-making, which is worth saving. It may be urged that it would cost too much to pump the water to correct this loss. In most cases the trouble is not in the quantity of water handled, but in the effectiveness of the apparatus; sometimes the water is wasted. This is especially true of atmospheric coolers, or where the pipes are badly scaled, forming an insulation, or where the surface is not sufficient for the work.

The economical apparatus is one that, by counter-current, makes practically a complete exchange of temperatures. That is, the distilled water leaving the cooler is at nearly the same temperature as the water used for cooling purposes, and on the other hand this cooling water leaves the cooler at nearly the temperature of the incoming distilled water.

For the distilled-water system there should be five thermometers—on the inlet and the outlet of cooling water, on the inlet and the outlet of distilled water, and on the outlet of the storage tank.

The ammonia condenser offers a similar problem to the distilled-water cooler. Here it is essential that the ammonia leaving the condenser be practically at the temperature of the cooling water. A thermometer should be placed in the liquid ammonia line leaving the condenser and in the water supply and waste lines. In some places it will be found that the ammonia, after leaving the condenser, rises in temperature before reaching the feed valve. It is therefore necessary that a thermometer be placed in the ammonia line near the feed valves. If the ammonia receiver and liquid lines are exposed to heat and cause a rise in temperature of the ammonia, they should be insulated.

Conserving Cold-Water Supply

In the cooling of both the distilled water and the liquid ammonia there is an opportunity, where only a limited amount of really cold water is available for cooling purposes, to use this water in connection with supplemental coolers for both the distilled water and the liquid ammonia, and after being used for this purpose it would still be available for makeup water for cooling towers.

The losses due to the heating of the ammonia liquid as well as to the superheating of the suction gas were gone

into quite fully in a previous article,[1] but how is an engineer to know the temperature of the ammonia or the superheat unless the liquid and suction lines be provided with thermometers?

It is gratifying to note that most of the manufacturers of refrigerating machinery are now equipping their compressors with thermometer wells on both the suction and the discharge. A number of articles have been written on the use of thermometers at these two points, and the engineer who has become accustomed to studying the condition of his compressor by the reading of these would sadly miss the information that they constantly give him as to the operating conditions.

It is to be regretted that in the early days of the industry engineers were not more accustomed to the use of thermometers. The attempt to operate the plant on gage pressures alone has taken such a firm hold that it has militated against the use of the so-called "flooded system," which is without doubt the true method for operating refrigerating plants.

All engineers are familiar with the use of thermometers for the boiler-feed water, and yet one frequently finds a plant where the engineers cannot tell you the temperature of the water going to the boiler; or, if they have a heater, how much useful work this is doing.

While in this article the writer has dwelt on the compression system of refrigeration, there is a similar need for the use of thermometers in the absorption. The object is not to outline every point where the use of thermometers is valuable, but to give such illustrations as will cause the engineer to think and act for himself.

[1] "Power," June 8, 1915.

Manufacturers' Instruction Cards Helpful

By F. W. Fisher

There seems to be a growing tendency among the better-class manufacturers of today to send out with their products suitable instruction cards for the use of the producer, that he may properly erect and maintain whatever machinery purchased—a radical change from the old-fashioned *"Caveat emptor."*

Machinery is often delivered to some remote locality where mechanics are scarce and a thorough description of the machine or instrument, with sectional cuts if possible, is of much value. Many excellent mechanical appliances have been discarded or replaced by others that are better understood and the reputation of the former injured by neglecting this feature, causing general dissatisfaction and as a rule breeding another knocker. Knocks travel farther and faster than praise, even if undeserved.

Having visited several so-called power plants in rural communities of the South, I can truthfully say that the manufacturer has a hard time of it, being unprotected after he has received his payment. Many plants are operated by men who have no idea of the principles of mechanics, whose sole interest in life is three meals a day and a place to sleep; so it is little wonder that things go wrong—more wonder that they survive.

In a certain sawmill the saw filer was also the master mechanic. He had employed a burly Ethiopian to heave in coal and shavings, carry lumber and "keep 'er iled." The saw filer started things in the morning and left the oiling to the coal heaver, who could not oil the governor while it was running; in fact, he did not know it needed occasional oiling, so that it gave trouble at times. When the engine began to race he would "choke her down" with a 6x6 oak timber brake. From the appearance of the groove in the timber it was frequently used.

A well-known maker shipped a pump to a mine in this district, but unfortunately failed to send instructions with it. The pump was duly installed, after which the engineer proceeded to "see what was inside of it." In taking the steam end apart the gasket was torn. A new one was made of very heavy gum packing, the pump reassembled and the steam turned on, but the piston made one stroke and died. The efforts of all the engineers (?) in the neighborhood were of no avail when it came to bringing it to life again, so the maker was wired to send a man from the factory. Upon his arrival he gave orders to remove the steam chest, but was told that this was unnecessary as they had tinkered with it several times and found it all right, but he insisted and immediately saw the cause of the trouble. Taking a penknife he cut a small hole in the packing, which opened the port leading to the auxiliary valve allowing the pump to work satisfactorily. Of course the mine operators had to pay a nice bill for the time and travel, which would not have occurred had suitable instructions gone with the pump. This make of pump was heartily knocked in that region after this unfortunate occurrence.

Such incidents are continually taking place, and they certainly do the manufacturer no good. Instructions are often sent out that never reach the parties for whom they were intended. This should be guarded against and a few dollars, judiciously spent, will do much to correct the evil and be of great benefit to both manufacturer and buyer. Instruction cards should be printed on one side only, so that they may be framed or tacked to walls near the machine. It would be better to forward two copies, one for immediate use and the other to be filed for future use. Many engineers have been helped out by retaining their sets of instructions. It might be well to label them "Do Not Lose," or "This Card is Valuable," rather than be open to ridicule. Many engineers will not ask information from someone using the same type of equipment. Instruction cards help here.

This subject is worthy of deep thought, and it is earnestly hoped that manufacturers will see their way clear to adopt methods for giving proper operating instructions. Textbooks are of very little value when installing or having trouble with some individual or special machine.

Pump and engine builders have been in the lead in this work in the past, but the electrical men are rapidly forging to the front and threatening to take the lead. When purchasing new machinery insist upon complete instructions as to erection, design and operation, and your troubles will be diminished. No engineer understands all machines and instruments and should be assisted in every way possible, and he will then become a booster.

Cotton Waste, which is a common acquaintance of well-kept engine rooms, is a sort of byproduct of spinning and weaving mills which is converted into what is known as wiping waste by passing through a special machine with a series of toothed rollers somewhat suggestive of our old-time wool-carding machines. It is, however, a much more complicated machine which originated in England.

Editorials

Educational and Reference Value of Advertisements

If all the information that is contained in, and deducible from, the advertisements which appear in *Power* in the course of a year were put into a correspondence course, hundreds of engineers would pay from twenty-five to fifty dollars for it and write glowing testimonials of the satisfaction and benefit derived from it.

The average reader has not learned to make the fullest use of the advertising section of his paper or to appreciate the difference between the advertising of today and that of a decade ago.

The old advertisement was comparable to the blare of trumpets and roll of drums which attract the unoccupied within the range of the street orator's eloquence, or to the vociferous superlatives of a Coney Island barker. Its object was to attract—it did not aim to instruct. The almost universal policy was to tell just enough to arouse curiosity and get the victim to nibbling.

Modern advertising proceeds upon the assumption that there are people who want and need the things advertised, that their patronage need not be begged as a favor, but will be gladly accorded when they are shown that the advertiser has what they need and can use to advantage.

These observations are suggested by a letter from one of the readers of *Power* containing two hints. The first is to go through each copy of the paper and describe to somebody, an interested listener if one can be found, to oneself mentally, or, better still, put into writing an account of each apparatus there found and describe how it works and what it is good for. One will be surprised to find how many little things he finds himself wondering about when he tries to write such a description and how much better he will know the subject of his article when he gets through.

Such an intimate collective knowledge of all the things that appear in *Power* would be some capital for a power-plant engineer. To know the latest and best that is available for all sorts of power-plant needs, the characteristics of different examples of the same type, that which is best adapted for each set of conditions, and their various prices would not only make him a master of his own department, but an authority upon the general subject.

The second hint was for the preservation of the information thus gotten together in a simple, easily accessible form. One cannot carry everything in one's head, and may as well be without a thing as to not know that he has it. It is therefore suggested that illustrations, tables of dimensions, special information of all kinds be clipped and pasted into scrapbooks or otherwise filed under appropriate headings. A filing drawer divided by consecutively numbered cards makes less work and is more flexible than a scrapbook. A small card index will give the number of the card in front of which certain information is filed, and all the material relating to that subject can come together in that place, which should be large enough to take in catalogs, etc. The names of specialists in, or books upon and formulas pertaining to, the subject may be entered upon the index card. A few years of this kind of collecting would put an engineer in possession of information that would be invaluable.

The Operating Engineer as a Business Man

Along with the study of technical and engineering subjects, the engineer of the present day, to attain more than mediocre success, must study business methods and become at the same time a competent engineer and a thorough business manager.

A suggestion pertaining to a change or improvement in the plant may be thoroughly sound from a mechanical standpoint alone, but when put to the test of a close analysis as a business proposition may prove to be a money loser. The successful bank president will not become enthused over a suggestion to put money into a project, however interesting, if it fails in the only test known to him—will it pay? Likewise the engineer, if shown a piece of apparatus, highly polished and beautifully finished, might naturally be expected to ask, Will it work? If the answer is negative in either case the deal, of course, is off.

Much can be learned from the methods of a successful salesman by noting his systematic progress. To receive attention at all he must, of course, present something which is applicable or usable in the particular plant visited, then show that it is preferable to others and why. He will probably depart after a short explanation in well-chosen words, leaving circulars or printed matter to be looked over at leisure, closing the first stage of the campaign. The engineer, on the other hand, gets a shadowy mental picture of what he would like to have for his plant, hails the "boss" at an inopportune time and perhaps, at long range, in an antagonistic or half-earnest, doubtful attitude, and when questioned as to first cost or cost of maintenance, benefits expressed in the only terms intelligible to the business man, he flounders miserably, considers himself affronted and loses the whole case. Could a salesman succeed by the same method?

On the other hand, how much more dignified the whole incident would be if Mr. Boss is called on the 'phone some day and told that Mr. Engineer has a business proposition—not in regard to an increase in wages—to present and would like an engagement with Mr. Boss in his office. If he gets the offhand reply, "Oh yes, any time when I am not too busy," Mr. Engineer should insist on a definite time being set on the score that he can talk better and feel more at home in the office when in a clean and presentable condition—a fact worth considering. When the time for the interview arrives Mr. Engineer is ready with a neatly prepared statement of the case giving all the known facts and figures, to be left with Mr. Boss, supplemented by such verbal explanations

as the case may require, and is prepared to answer questions in an intelligent manner.

The comparison is by no means overdrawn, nor is it difficult to foresee the difference in the results to be obtained by the two processes.

National Water-Power Legislation

Water-power conferences and conventions come and go; in fact, like the streams they seem to go on forever, without actually accomplishing anything as far as the actual use of the water powers is concerned. As recently as November, 1914, at the seventh annual convention of governors, held at Madison, Wis., the Western governors came out strongly in favor of allowing the separate states to control and to regulate the water powers within their borders. They were opposed to any leasing system by which title to the water powers would be vested in the National Government. This position of the governors, as evidenced by the opposition of some of the Western senators, resulted in the defeat of the Administration water-power legislation proposed at the last session of Congress.

The Western governors met at Seattle last May, and there was then some support of Secretary Lane's plan for the leasing of water sites on the public domain. This plan was incorporated in the Ferris Bill, which was defeated in the Senate last winter. The Seattle conference did not favor any definite legislative program, as it was the sentiment to postpone action until the water-power conference which the governors are now (Sept. 21-23) attending at Portland, Ore. Meanwhile Senator Chamberlin of Oregon, in an address delivered recently before the legal associations of Washington and Oregon, admitted that under present conditions the development of water power is an impossibility, but he strongly advocated the passing of the Ferris Bill, which would safeguard the public and at the same time invite the investment of capital in power development. The closing paragraph of the address is well worth quoting:

> I appeal to each of you, as loyal and patriotic citizens, to lend your efforts to the enactment of a law which, while it will assist in the development of the West, for the enjoyment and enrichment of the present generation, will at the same time protect them against extortion and place it in the power of future generations to control and regulate a business which may be of greater importance to them than to those of the present.

Considerable opposition to the Ferris Bill was shown at the annual conference of governors held at Boston in August. On this occasion Governor Lister of Washington and ex-Governor Hawley of Idaho spoke against its passage and favored state control of water-power.

Whatever action the Portland conference may take, the essentials of the Ferris Bill should be passed at the coming session of Congress. It is quite likely that after it is made a law many of the interests now opposing the bill on the ground that its provisions are inimical to capital will find that it offers reasonable protection to the public and that it also provides attractive conditions for investors.

If the bill is to be passed at the next session of Congress, steps must be taken to arouse public opinion, and through public opinion the members of the National Legislature, to the necessity of immediate action. Otherwise the specious plea of states' rights will again be used to defeat the measure.

With few exceptions the Western water powers not owned by the Federal Government have passed into private ownership in perpetuity. In justice to the present and to the future generation the national resources owned by the public should be conserved to give the greatest good to the greatest number.

What Obsolescence Is

Judging from letters from, and conversations with many operating engineers, there seems yet to be some misunderstanding of the term obsolescence. According to Webster the word means ancient, antiquated, old-fashioned, antique, disused, outworn, neglected. As applied to power engineering, obsolescence is not adequately defined by any of these.

Whether or not a particular piece of power-plant apparatus is obsolete depends not altogether upon the fact that some other design or kind of machinery for the same purpose has been newly made. The reciprocating engine as a class of prime mover is not obsolete because the turbine is a more recent development. In engineering a piece of equipment is obsolete only when the total advantages, based chiefly on dollars and cents, of more recent designs outweigh those of older designs. It is evident then that service is a factor in determining if an apparatus is obsolete.

In one kind of service, equipment might become obsolete in a few years, while in another it would never reach that state. The great reciprocating engines, though in excellent condition physically, were obsolete for the Seventy-fourth Street Station of the Interborough Rapid Transit Co. because the complete expansion turbine could meet the service demands at less total cost per unit of output than the engines were capable of. For nearly all services employing units of large capacity, it may be said that the reciprocating engine is obsolete. The condensing turbine has developed remarkably high efficiencies, requires comparatively little labor and supplies, and occupies less floor space per unit of capacity than the reciprocating engine. But take the case of a plant where cheap condensing water is not available and where all or nearly all exhaust steam can be utilized. The turbine, condensing or noncondensing, would find it difficult to compete with many types of reciprocating engines.

In engineering obsolescence is relative, like wealth, time, distance and market values.

The man who is particular, even "cranky," about his wrenches is generally neat about his work.

The cub engineer who thinks it makes him look better qualified to get good and greasy all over has another think coming.

Twenty grimy men of the engine-room crew of the "Arabic" were the heroes of yesterday's tragedy, according to Captain Finch, of the liner.

"These brave fellows remained below, hoping to be of service, though they might have escaped," said Captain Finch. "Every man of them was lost. They deserve 20 Victoria Crosses. They're real heroes—a thousand times over."—Evening Mail.

"Hoping to be of service." That's all.

Correspondence

Boiler Tests at Various Loads

Mr. Rogers, in his article, "Boiler Tests at Various Loads," of July 20, states that his tests "corroborate the wisdom of present-day practice of carrying overloaded boilers to get better efficiency," and that at 200 per cent. of the rated load efficiency was no lower than at 115 per cent. This is contradicted by his own curve drawn from results of his tests, showing a peak of the curve indicating highest efficiency at between 110 and 120 per cent. of the rated load.

Mr. Maynz's criticism of Mr. Rogers tests in *Power* of Aug. 17 is also faulty. His opinion that 0.17 and 0.18 in. of draft is too strong to burn 14.3 and 15 lb. of coal per sq.ft. of grate surface per hr. is wrong, as the draft above the fire depends upon a number of circumstances—thickness of the fuel bed, condition of the fire and the amount of fine matter in the fuel. To illustrate, we can take the following resulting figures of a number of boiler tests, made in all four cases with nearly the same kind of coal:

Horsepower Developed	Fuel Thickness	Draft Required in Furnace
500	6 in.	0.17 in.
	7 in.	0.23 in.
	8 in.	0.29 in.
	9 in.	0.45 in.

These figures are, of course, subject to change under the conditions outlined; but nevertheless one concludes from them that by maintaining the same draft in the furnace and by increasing the fuel-bed, the amount of air passing through will be reduced, and consequently the coal burned and steam generated will be less. By decreasing the fuel thickness the resistance to the air passing through it will be reduced, with the consequent greater flow of air, greater quantity of coal burned and more steam generated. If Mr. Rogers, when working the boiler below rating, increased the thickness of the fuel-bed, he would have done best, as it would have decreased the air leakage through the fuel-bed, but at the same time would increase the draft over the fire. He should be able to obtain 10 to 20 per cent. overload with 0.15 in. of draft in the furnace.

The comparatively high furnace temperature with Mr. Rogers' under-rating is chiefly due to the sluggish movement of the gas over the heating surface. The absorption of heat in a boiler depends primarily upon the difference of temperatures, and secondarily upon the velocity of the gas over the heating surface. If the velocity is high the friction between the gas particles will be greater and the thickness of the dead film of gas next to the metal, which tends to retard heat transmission, will be less. This is true when velocity is increased without increasing the volume.

The actual ash obtained from a furnace is less and often considerably less than the amount computed from the quantity of coal. This reduces the value of Mr. Maynz's method of obtaining the quantity of fixed carbon in ash, and if Mr. Rogers would have had the ash analyzed the amount of carbon in it would probably be much lower than the amount stated by Mr. Maynz.

The efficiencies obtained in Mr. Rogers' last three tests are ordinary ones obtained by me with water-tube boilers at corresponding overratings. Tests conducted under my supervision upon a 250-hp. Stirling boiler gave an efficiency of 72 per cent. when operated with a load of between 100 and 110 per cent. of its rated capacity, and this while the boiler had a ratio of heating to grate surface of 55.5 to 1 while Mr. Rogers' boiler has a ratio of 62.7 to 1. Under other conditions and fuel I obtained an efficiency of 76 per cent. with the same boiler, and this was often obtained before with this size of this type of boiler. The foregoing disproves the opinion that small boilers are more economical than large ones as long as fuel economy only is concerned. This is also proved by Mr. Rogers' boiler when working at underloads—the tests indicate an extremely low efficiency. In small power plants it is more economical to have two or four small boilers than one or two large ones; because, if the load drops off when two boilers are in operation, one could be banked, burning little coal, and the other operated at its most economical rating. In addition to this in case of breakdown of a boiler or stoker many plants would be crippled, and smaller ones would possibly have to shut down.

VICTOR J. AZBE.

St. Louis, Mo.

Bleeder Turbine Runs Away as Low-Pressure Unit

A rather peculiar case of a turbine running away occurred recently in a small central station in Ohio. The machine was a 1,000-kw. bleeder type, running at 2,400 r.p.m. Steam was taken off at the bleeder connection at about fourteen pounds above atmosphere to supply a town heating system. Between the bleeder connection and the heating main was a gate valve and a horizontal swing check valve. Ordinarily, this one unit handled the whole load. The reserve consisted of a 400-hp. Corliss engine exhausting directly into the heating main.

At the time the accident occurred the turbine was being shut down to inspect the governor. The reciprocating engine was put on during a light-load period, the load transferred to it and the turbine taken off the board. At once the turbine began to overspeed. The operator, who was new at the plant, hurried to shut the throttle, thinking the automatic had failed to work. Instead of slowing down however, the machine continued to pick up speed. It was now running at a dangerous rate and throwing out fire at several points where the spindle was striking. The operator even climbed up and shut off the valve from the steam header to the turbine, but it had no effect. Then he ran around and shut the valve in the bleeder connection. This proved to be the source

of the trouble and the turbine began to drop in speed; but not until considerable damage had been done.

It was subsequently found that the check valve in the bleeder line was broken. And as soon as the turbine was off the board running light, steam started to flow from the heating main back through the bleeder connection and into the turbine and condenser. The turbine was running as a low-pressure machine with the governor having no control. The machine was out of service three or four weeks and the repairs cost several thousand dollars.

D. N. McClinton.

Pittsburgh, Penn.

Small vs. Large 550-Volt Armatures

The question is often asked, Why do small direct-current motors give more trouble than larger machines of a corresponding voltage? This is very noticeable among the 550-volt direct-current class.

In the first place, manufacturers cannot build the smaller motors with the same resistance to abuse as the large station generator units of the same voltage. Usually the armatures of these direct-current machines give most of the trouble.

For example, consider a 550-volt, 4-amp., 2-pole motor and a 550-volt, 1500-amp., 10-pole motor. The former has, say, 80 segments or two circuits of 40 segments each with 2 amp. per circuit and an average cross-segment potential of 13.75 volts. Should any metal dust become embedded between two segments, or the mica partly carbonize, it would prove an easy path for 2 amp. to flow and at the same time short-circuit the armature coil. This process usually continues until the coil is cooked and all the turns are shorted, since there are many turns of small wire in small machines. It is the writer's experience that more armature coils of small 550-volt machines are burned out by this cause than by any other; with the possible exception of a low-grade mica between segments.

Such burnouts could hardly happen to large armatures from such minor causes. Therefore, it is imperative to anyone looking after small direct-current motors to keep the commutator, brushes, and brush-holders scrupulously clean if they wish to invite minimum trouble.

The larger motor has, say, 800 segments, or 10 circuits of 80 segments each, with 150 amp. per circuit, and an average cross-segment potential of 6.9 volts. It is evident that the particles of metal, or the carbonized mica mentioned in the former case, would be quickly burned off by reason of the larger amperage. There is less cross-segment potential and less chance to short-circuit in lieu of the regular armature path. In the former case there is a higher cross-segment potential, with far lower amperage. Moreover, small motors have armatures wound with small wires, especially the 550-volt type; hence when they flash at the commutator due to obstruction, dirt, etc., they very often sear off one or two armature leads.

Small motor armature commutators, being small, are easily grounded by reason of smaller insulation spacing and less distance to grounded parts of the machine. Also the armature leads break loose easily, causing an open circuit, due mainly to continual expansion, contraction and vibration. Large motors have heavier leads, hence stand up better.

The writer has not attempted to give all the troubles that come to direct-current armatures, but merely states the principal causes and weaknesses of the small direct-current armature compared with the larger units of the same voltage design.

Ben Dawson.

Cedar Rapids, Iowa.

Tests of Firebrick Linings

I believe it would be of interest to the readers of *Power* who have had difficulty with boiler brickwork when forcing boilers at high overloads to find out what brick would give the best results. I would like to make a suggestion that some boiler-room engineer try the following experiment:

Line one side of the furnace with bauxite brick at the clinker line, the other side with standard firebrick of the grade he is using at present. In another furnace line one side with silica brick and the other with standard firebrick. With furnaces lined thus, make a complete list of results as to the length of service, cost of the brick, rating of the stoker, peak and average loads, the kind and analysis of the coal used, and the general characteristics of the coal and the ash.

The results would be interesting, especially in regard to the action of a basic or acid ash on the silica and bauxite brick, and should any engineer try this experiment and publish in your paper the complete results, he will have rendered a needed service to engineering.

Theodore Maynz,
Engineer, Combustion Engineering Corporation.
New York City.

"Slide-Rule Quadratics"

In solving for roots of quadratic equations the slide rule is quite valuable. Not only is the process accurate (as far as slide-rule mathematics can be considered accurate), but the results are reached quicker (and with perhaps less mental energy) than by substituting in the formula,

$$x = \frac{-p \pm \sqrt{p^2 - 4q}}{2}$$

The rational basis of slide-rule quadratics is as follows: Every quadratic can be expressed as

$$x^2 + px + q = O$$

in which the coefficient $-p$ equals the sum of the two roots and q is their product.

(1) When both roots are *positive* and equal to a and b, then

$$x^2 - (a + b)x + ab = O = x^2 + px + q$$

and therefore $p = -(a + b)$ and $q = ab$, or if $\frac{q}{a}$ is substituted for b then $p = -\left(a + \frac{q}{a}\right)$.

(2) When both roots are negative p is of opposite sign to that given in (1).

(3) When the roots are of opposite signs and a greater than b, the coefficient $p = \pm(a - b)$ and $q = -ab$ and therefore $p = \pm\left(a + \frac{q}{a}\right)$; that is, it is numerically dependent upon the sign of the roots.

The slide rule can be used to solve for the numerical value of roots a and b. Since $p = \pm\left(a + \frac{q}{a}\right)$, by setting the runner of the slide rule on q and shifting the slide the quotient $\frac{q}{a}$ can be found, a being any trial divisor; then, mentally the trial divisor a is added to the quotient $\frac{q}{a}$ so as to give a sum equal to p.

Example—Take the quadratic $x^2 + 75x + 15 = O$. Here $p = 75$, $q = 15$ and $p = a + \frac{15}{a} = 75$. By setting the runner at 15 and shifting the slide, we find that when the trial divisor a reaches a value of about 0.2005 the quotient $\frac{q}{a}$ becomes 74.80, which, when added to the trial divisor a, gives a sum approximately equal to 75, the value of p. Likewise roots of $x^2 - 75x - 92 = O$ are —1.208 and + 76.208.

Not only has the trial setting enabled us to check the sum of the two roots, but also the value of the second root has been found directly from the quotient $\frac{q}{a}$.

By the foregoing method with the aid of a little mental arithmetic quadratics may be readily solved, and the advantage becomes more apparent as the character and values of p and q vary.

ROBERT N. MILLER.

Pittsburgh, Penn.

Direct-Reading Barometer

Regarding the direct-reading barometer described by Miles Sampson on page 164, Aug. 3, I wish to ask what method he uses to fill a tube made in this manner?

It is very difficult to fill a straight tube and get all the air out of it, and it seems to me that it would be next to impossible to fill with mercury a U-tube having one end closed and get a perfect vacuum, with the facilities at hand in the engine room. It would be interesting to have published the method of filling such tubes.

J. E. PELTON.

Portland, Ore.

Broadening Responsibilities of the Engineer

The editorial in the July 27 issue, "Broadening Responsibilities of the Engineer," interested me and recalled a little experience I had along this line a few years ago. I was employed by a large manufacturing company with the understanding that in a few weeks I would be placed in charge as chief engineer, but things seemed to drag along and this authority was not invested in me.

Finally I went to the superintendent and called his attention to our agreement. He said that it was his intention to do just as he agreed, but there was a stumbling block which he had not considered, and he hardly knew just how to get by it. There were at two of the company's plants two old men who had enjoyed the distinction of being called "chief engineer," and he was afraid that if this title was taken from them it would cause friction and discord at the plants.

I suggested, as a way out of this difficulty, that he install me in the plant as superintendent of motive power and let those men still be called chief engineers at their respective plants. This was done without any hard feeling whatever. These men performed their duties as before under the same title, only they reported to and worked under me instead of the superintendent, as they had done previously. I believe that in large plants the title superintendent of motive power is much better than chief engineer.

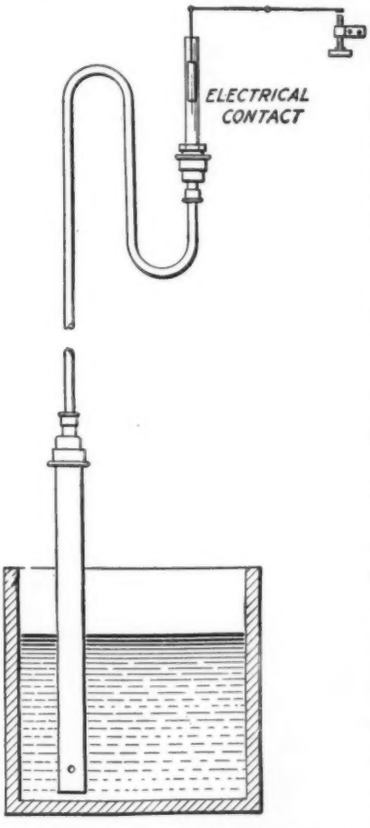
H. R. ROCKWELL.

St. Louis, Mo.

A Simple Water-Level Indicator

On visiting one of our local power plants recently I was interested in a simple and efficient gage that was used to show the level of the water in the hotwell.

This plant was equipped with a jet condenser discharging into a trench in the basement floor and then into the well outside of the building. The drain leading from this well to the sewer is barely large enough to handle the necessary injection water, and if too much water is turned on it will overflow into the basement and the boiler room. To avoid this trouble, the usual float, cord and pulley combinations were tried, but proved unsuccessful. The result of the next experiment was a pronounced success. A piece of three-inch pipe about 3 ft. long, with a ⅜-in. hole drilled in the side of it, just above the lowest water level, was set in the well, with the lower end just clear of the bottom. On the upper end of this pipe was a reducing coupling and a ¼-in. pipe led upstairs into the engine room and was formed into a U-shape and mounted on the wall. The final vertical side of the U was made of a gage-glass set into a stuffing-box made out of a reducing coupling and a bushing and was partly filled with oil.



INDICATOR AS IT MAY BE CONNECTED

The operation of the gage is as follows: When the condenser is shut down, and the water in the well is at its lowest level, the oil in the U-tube just shows in the glass. When the condenser is in operation, the water rises in the well and the pipe, compressing the air in the 3-in. pipe, and in rising and falling causes a corresponding rise and fall in the U-tube, because of the varying air pressure in the pipe. Each time the condenser is stopped and the well emptied the hole in the side of the well pipe

is uncovered or opened to the air, allowing the gage to refill and adjust itself automatically. The ratio of rise and fall between the oil in the tube and the water in the well can be varied by changing the ratio between the diameter of the well pipe and the diameter of the gage glass. This gage is so sensitive that the surface of the oil vibrates from a slight agitation of the water in the well.

To further insure against a flood, an electric alarm, consisting of a float to operate a flat metal lever pivoted in the middle, is mounted on the top of the gage glass. One end of the lever is fastened to the float or piston in the tube, while under the other end is fitted a contact screw set in a piece of wood for insulation. One wire of the bell circuit is fastened to the metal case, the other to the contact screw. When the oil in the glass reaches the high limit the piston rises lifting the end of the lever and bringing the other end in contact with the screw, which closes the circuit and rings the bell. Reducing the injection water reverses the operation and stops the ringing. The float was made by using a small glass vial which fitted the tube quite loosely. When the oil reached the bottom of the vial, capillary attraction caused the oil to fill the space between the vial and tube, thus supplying the necessary packing. A gage of this kind could be adapted to a number of different uses.

H. K. Wilson.

New Bedford, Mass.

Boiler-Feed Pump Discussion

Referring to the discussion on centrifugal boiler-feed pumps, started in the issue of July 27, page 132, by C. G. Richardson, the centrifugal pump has all of the advantages mentioned by him when compared with the direct-acting steam pump. It is probably true that in selecting boiler-feed pumps or other auxiliaries, as well as in selecting the major equipment, the possible arrangements and combinations of apparatus should be thoroughly investigated from an impartial engineering standpoint. Engineers are apt to be too enthusiastic over apparatus that has been purchased at their recommendation. It is also too often the case that they are intolerant of suggestions that something else might serve the purpose better.

The inefficient direct-acting steam pump has maintained its supremacy as a boiler feeder, for the following reasons: Its low first cost; reliability (its poor economy is not so noticeable as its reliability even when badly abused); and the inclination of engineers to follow precedent, or as Mr. Sandstrom in the Aug. 24 issue calls it, "mental inertia." Some contend that efficient auxiliaries are superfluous because large quantities of the waste heat may be reclaimed by the use of feed-water heaters, and this is probably the principal reason. My observation indicates that a higher overall thermal efficiency may be obtained by using highly efficient auxiliaries and using the feed water to abstract P.t.u. from the flue gases by the use of economizers or in the boilers direct. Other considerations being equal, the pump that gives the average water horsepower for the least expenditure for coal is the one to be desired, therefore the great difference in the steam consumption of direct-acting pumps and centrifugals, in the larger sizes, eliminates the former from competition on this basis. Some contend that "the triplex pump is unsuitable for boiler feeding," but is it, or is the statement made because its high efficiency is not readily apparent?

The triplex plunger pump has in its favor that it gives a steady flow of water, the cost of power is very much less than the centrifugal pump when applied to boiler feeding, it may be automatically regulated, it is reliable and if given intelligent attention it will maintain its high efficiency for 15 to 20 years, as has been demonstrated, with no cost for repairs except for packing and valves.

The high efficiency of the triplex pump is attained not merely at its rated capacity, but is nearly constant throughout the full range of operation provided its capacity is regulated by changing the speed. The average efficiency is therefore greater than a mere comparison of catalog percentages would indicate.

The triplex pump has a practically constant efficiency at various speeds. The capacity is proportional to the speed. The discharge head does not have to be throttled to regulate its capacity. The efficiency of a variable-speed direct-current motor such as is used to drive triplex pumps is more nearly constant at variable load and speed than the efficiency of constant-speed motors at variable load used to drive centrifugals, and small reciprocating engines have much better efficiencies at variable speeds than small turbines at variable loads.

Comparing two types of boiler-feeding units, one a motor-driven centrifugal pump and the other a motor-driven triplex pump, taking into consideration the daily load curve of the plant and the efficiency curves of the two pumps, together with the efficiency curves of the two motors, it was found that the actual coal required by the triplex pump would be less than one-half that required by the centrifugal. A similar comparison covering steam-driven units would show even greater difference in favor of the triplex pump. Against these advantages are, more space required, higher first cost, more complicated apparatus and more attendance. But these are much more than compensated for by the higher economy in the case mentioned. I therefore often wonder why so few plants use them for boiler feeding.

Burt N. Everett.

Jamestown, N. Y.

Peripheral Speeds

On page 26, July 6, reference is made to a paragraph appearing in *Power,* June 15, p. 815, in which it is stated that 38 ft. per sec. is considered the limit in safe speed for a cast-iron flywheel, with the inference that this figure was given by the undersigned in a paper read before the Manchester Association of Engineers. Wherever this figure came from, it certainly did not come from the paper referred to, which contained no reference whatever to cast-iron flywheels or safe speeds for them.

R. F. Halliwell.

Rugby, Eng.

[While the item in the issue of June 15 properly quoted the part taken from Mr. Halliwell's paper, a mistake was made in the editorial note on page 26 in crediting it entirely to him, whereas the first part was evidently supplied by the writer of the item.—Editor.]

Inquiries of General Interest

Allowable Variation in Voltage—What is the usual voltage variation allowed in electric systems?

J. C. H.

A variation of 5 per cent. above or below the normal voltage is customary for power circuits, and for lighting systems about 2½ per cent. above or below the normal voltage.

Leakage of Open Heater—What would cause sudden leakage at the joints of a cast-iron open heater?

M. L. M.

Leakage could be produced by unequal expansion or contraction resulting from a sudden change of temperature, as warming up the heater with exhaust steam when the heater was empty or filling the heater with cold water after it had been warmed by the exhaust.

Source of Air Supply for Indirect Heating—For an indirect system of heating, in place of warming cold air taken by a duct directly from out-of-doors, would it not be more economical to take the air from another room?

D. S.

There would be greater economy of heat, but unless the air supply was constantly replenished by outdoor air, the system would be little better than heating by direct radiators and the purpose of indirect heating, which is to obtain a supply of heated fresh air and thereby secure ventilation as well as warmth, would be defeated.

Testing a Gas Meter—How can a gas meter used for measuring natural gas be tested to ascertain whether it registers correctly?

W. L. T.

The usual method of testing gas meters is to compare their registration with that of a standard meter by passing the same gas at the same pressure through both meters; or by checking the registration of the meter under test with the volume discharged by it through a calibrated aperture; also by comparing the registration with the volume of gas discharged through the meter in filling or emptying a gasometer of known dimensions.

Friction Test of Rubber Belting—What is meant by friction test of rubber hose or rubber belting?

C. R. H.

In rubber hose, rubber belting and other articles made up with superimposed layers of canvas, the "friction" is the soft-rubber compound which is applied to the canvas by means of which the different layers or plies, are held together. The rubber layers are probably called "friction" because the rubber is applied by passing the canvas between friction rolls, thus causing a wiping action which forces the "friction" material into the meshes of the canvas. A friction test consists in ascertaining the adhesiveness of the canvas by observing the rates at which a given width becomes stripped from the "friction" when subjected to a stated pull in pounds.

Suction Head of Centrifugal Handling Hot Water—For operation of our centrifugal pump, why is it necessary for the suction water at 212 deg. F. to be supplied under a head of about 8 ft.?

R. C. D.

To supply the pump, the suction water must be urged by sufficient pressure to overcome the pressure of air or vapor liberated on the suction side of the pump and also for overcoming the suction lift and friction of the water in the suction pipe and passages. Having a temperature of 212 deg. F. and pressure at or below that of the atmosphere, the suction water would vaporize, causing the pump to become filled with steam at atmospheric pressure that would prevent the admission of water. To obviate such a formation of vapor and overcome the frictional resistances, the suction water must be supplied under a pressure head greater than the pressure of the atmosphere.

Individual vs. Group Driving—What is the relative economy of driving machinery by individual electric motors as compared with group driving?

J. S. B.

By use of individual motors, mechanical transmission losses are confined to those incident to the individual machines, while in group driving transmission losses continue practically the same whether all or only a portion of the machines of a group are in use. Hence, where a considerable proportion of the machines of a group may stand idle or are of different operating characteristics individual drives will be more economical, besides dispensing with the presence of transmission machinery. But where the machines of a group are close together, are of the same type and are run continuously, then with ordinary methods of transmission, group driving will usually be more economical on account of the higher efficiency of a larger motor and lower line losses.

Working Pressure for Stay-Bolted Surface—What pressure per square inch would be allowable for a flat boiler surface of $\frac{7}{16}$-in. steel with stay-bolts pitched 8½x8½ in.?

R. C. A.

The maximum safe working pressure would be given by the formula

$$P = \frac{C(t + 1)^2}{S^2 - 6}$$

in which

P = working pressure in lb. per sq.in.;
C = a constant = 66;
t = thickness of plate in sixteenths of an inch;
S = pitch of stay-bolts in inches.

Substituting t = 7 and S = 8.5, the allowable working pressure is found to be

$$P = \frac{66 \times (7 + 1)^2}{(8.5)^2 - 6} = 63.75 \text{ lb. per sq.in.}$$

Strength of Double-Riveted Lap Joint—What would be the efficiency of a double-riveted lap joint made of $\frac{7}{16}$-in. plate of 55,000 lb. tensile strength, with rivets $\frac{13}{16}$ in. dia. after driving and pitch of rivets 2.708 in.?

F. C. M.

Per unit of pitch

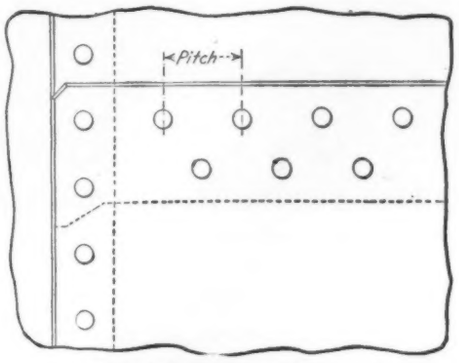
(A) The strength of solid plate would be

$$2.708 \times \tfrac{7}{16} \times 55{,}000 = 65{,}161 \text{ lb.; and}$$

(B) The strength of plate between rivet holes would be

$$(2.708 - \tfrac{13}{16}) \times \tfrac{7}{16} \times 55{,}000 = 45{,}610 \text{ lb.}$$

Allowing the shearing strength of the rivets in single shear to be 42,000 lb. per sq.in. of cross-sectional area, then



DOUBLE-RIVETED LAP JOINT

(C) the shearing strength of two (2) rivets in single shear would be

$$2 \times 42{,}000 \ (\tfrac{13}{16} \times \tfrac{13}{16} \times 0.7854) = 43{,}553 \text{ lb.}$$

and allowing the crushing strength of plate to be 95,000 lb. per. sq.in. of cross-section, then

(D) the crushing strength of plate in front of two (2) rivets would be

$$2 \times \tfrac{13}{16} \times \tfrac{7}{16} \times 95{,}000 = 67{,}539 \text{ lb.}$$

As the strength of the joint would depend on (C) the shearing strength of the rivets, then the efficiency of the joint would be

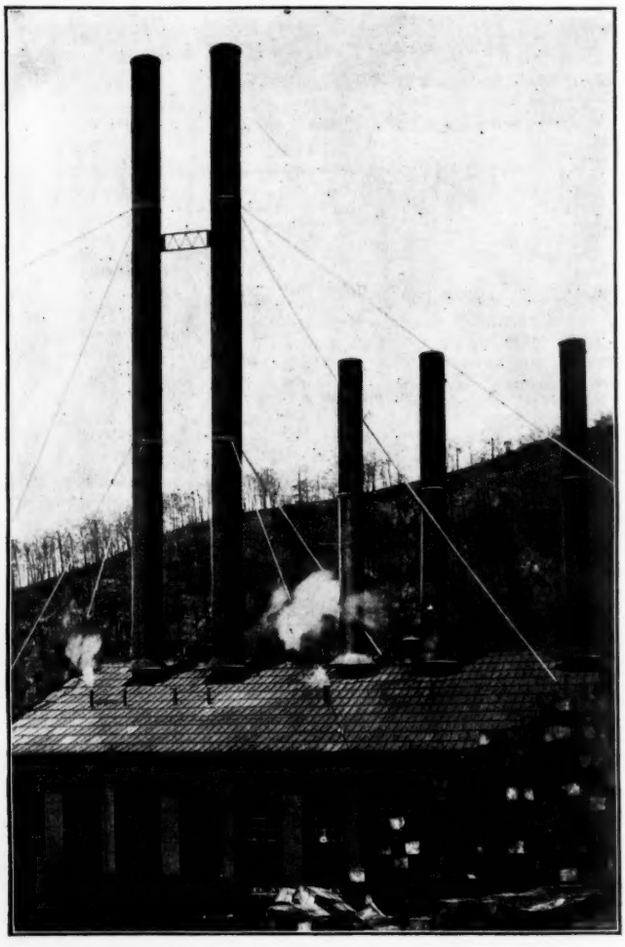
$$\frac{(C)}{(A)} = \frac{43{,}553}{65{,}161} = 66.8 \text{ per cent.}$$

[Correspondents sending us inquiries should sign their communications with full names and post-office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Wind Pulls Rivets in Stack Brace

SYNOPSIS—High wind caused rivets of brace to pull out of one of the two 100-ft. high steel stacks. A stiffening plate was placed on the inside of each to prevent a repetition of the occurrence.

The method of bracing steel smoke-stacks, as illustrated, is common practice, but there is danger that the brace between the two stacks may pull out of either one



SHOWING BRACE BETWEEN STEEL STACKS, THE RIVETS OF WHICH PULLED OUT AT ONE END

or the other when resisting high wind pressures. With the guy lines as shown, those on one stack oppose those on the other, the center brace acting as a tie. Should the brace fail, either one or both of the stacks are in danger of falling. That this did not occur at one particular plant in the mountainous region of Pennsylvania was due more to good fortune than to the brace or guy wires.

This particular plant has several steel unlined stacks. The two tallest are 100 ft. high and 54 in. diameter. Each stack is braced by four guy wires, as shown, and they are stayed between by a built-up piece of angle iron and diagonal braces. The ends of the brace were riveted to the stack by four rivets on each side of the tee-iron forming the end of the brace, or eight rivets in each stack. This arrangement appeared to be secure enough to withstand the force of the high winds which frequently swept down the valley between the surrounding high hills where the power plant is located.

A few months ago a severe windstorm occurred, blowing down the valley, from left to right of the illustration, for a period of about thirty minutes, apparently doing no damage. The next morning when the chief engineer came on duty, he noticed that the brace had pulled away from the right stack about three inches. An examination disclosed that the rivets had pulled out of the stack. Probably this occurred at the final gust of wind on the preceding day, otherwise the stack would have been blown over on the roof of the boiler house, resulting in damage that would be difficult to foretell.

To prevent a recurrence, a stiffening plate was placed on the inside of each stack and the brace riveted to it, thus giving a greater area to carry the strain set up by high wind pressures.

Producing Magnesia Pipe and Boiler Covering*

Magnesia for pipe and boiler covering is made from limestone or dolomite containing magnesium carbonate. One firm has rock of this kind containing about 40 per cent. magnesia. The works are generally located at the deposit, to save freight, and a cheap method of quarrying and conveying the rock is devised. The native rock is a mixture of calcium and magnesium carbonates. Since the calcium must be removed, the rock is first calcined, using charcoal as fuel. The result is a combined oxide of calcium and magnesium. The carbonate has been changed to the oxide for both elements. The carbonic acid produced is used later. The calcined rock is next mixed with water in special rotating vessels where the lime is slaked. All impurities remain in the vessel, and all the lime is carried away by the overflowing water during the operation. The oxides are now in a state of suspension in the water, the percentage of soluble material being very small. The later processes consist in a juggling of "states of suspension" and "of solution" to separate the calcium and the magnesia.

The liquid is pumped into vertical metal cylinders about 5 ft. in diameter and 15 ft. high. These have pipes leading from the top head to near the bottom for the entrance of the CO_2, derived from the calcining. This gas is pumped from the calcining kilns and returned to these cylinders under a pressure of about 70 lb. It rises through the liquid, being absorbed by the oxides in suspension in the water, and reconverts them into carbonates. But this operation produces an important difference in the condition of the two carbonates —after the operation the calcium carbonate is in suspension and the magnesium carbonate is in solution.

The next step, the separation of the material, is achieved by a system of filtration. The liquid enters the interior of a series of wood cells perforated by small holes and lined with cotton canvas or cloth, which act as filters. The water passes out, carrying with it the magnesium carbonate, while the calcium carbonate in suspension is caught in the meshes of the cloth as a paste. Frequently the cell has to be opened and the cloth stripped from the frames for the removal of the calcium carbonate. The cloths after cleaning are again used for filtering. This is the end of the calcium carbonate as far as this process is concerned.

The water containing the weak solution of magnesium carbonate is next put into other vertical cylinders of about the same size as the carbonating cylinder and is there heated by steam to about 190 deg. F. At this temperature the magnesium carbonate goes again into suspension instead of into solution. The liquid is then pumped to open vats lined with thin cloth, and most of the water is drained off, just enough remaining to allow the mixture to be pumped to the molds.

The molds are like the filtering cells, with the same size of holes and kind of cloth. A number of cells are placed

*From an article by Captain F. W. Bartlett, in the August "Journal of the American Society of Naval Engineers."

beside one another and pressed together firmly, and the liquid is allowed to enter them all at once. When all are filled so that no more liquid flows from the openings, the cells are opened. The material that was in suspension has been retained by the filtering process, and the mold is filled with magnesium carbonate.

The magnesium carbonate used for pipe or boiler covering is mixed with asbestos fiber so that the resulting mixture may not be so brittle. The covering is formed in molds that resemble the kilns previously described, except that cores are made as in foundry work and the outside corresponds to the shape desired. After the molds are made they are dried for six days in extensive ovens, and later are finished just as if they were made of wood. Planers doing exactly the kind of work done by wood planers shape the outsides and insides. The material is cut to exact length by saws. Later, thin canvas is pasted on the pipe and valve coverings, and the product is ready for shipment.

Seattle Considers Purchasing Electrical Energy

Since the new $1,000,000 dam of the Seattle City Water and Power System at Cedar River practically failed, the system has been hard-pressed for power. Several offers have been made to sell power to the municipal plant, and also several power sites have been offered to the city. The Board of Works is now considering a proposition to seal the basin of the Cedar River reservoir. The north bank of the basin formed by the big dam is so porous that the improvement is useless in its present condition. J. D. Ross, superintendent of the City of Seattle lighting plant, in a communication to the City Council, stated in part:

During this summer (1915) our total cost, at the steam plant, including interest, depreciation, operation, and fuel, based on 80c. oil was 0.65c. per kw.-hr. The fuel cost was 0.45c. per kw.-hr. These figures cover a period when the fixed cost was distributed over a very large output, while Cedar River was practically shut down during betterment. The demand during the coming winter will be about 13,500 kw., leaving approximately 4,000 kw. to be supplied from steam or other sources.

Mr. Ross intends to get 1,500 kw. from the Lake Union hydro-electric plant whenever water is available. This will leave about 2,500 kw. to be supplied from the steam auxiliary plant or from other sources. A fourth boiler is being added to the steam plant. This will give one-third more capacity, and the cost per kilowatt hour on normal days would be about 0.6c. When the plant is used only for peak, emergency and stand-by service, the output is small and fixed costs must be distributed over a smaller output, thereby raising the cost per kilowatt hour. The fuel cost per kilowatt hour, however, remains nearly the same, being 0.5c. when the 10,000-kw. generator is operated at the lower-efficiency partial load. As the machine approaches full load the fuel cost will be 0.326c. The interest charges on the steam plant must be paid until the bonds are retired. Of the $425,000 issue, $21,000 has been retired, so that the interest charge is rapidly decreasing. The existing plant depreciates whether used or not, and this item, like that of interest, could not be lessened by purchasing power from other sources.

Large Battery of Cooling Towers—A large battery of forced-draft cooling towers, built by the Wheeler Condenser & Engineering Co., of Carteret, N. J., for the Texas Power & Light Co., Waco, Texas, are of interest. They are of the steel-shell type, arranged in a battery 100 ft. long, 40 ft. wide and 40 ft. high. There are six pairs of 10-ft. fans operating at 250 r.p.m., belt-driven by motors located above each pair of fans in small motor houses. The fan housings are extended and are provided with doors which may be opened to permit unobstructed entrance of air for operation by natural draft during the winter season. A platform reached by a ladder from the ground serves the three motor houses, and an upper gallery is built on the level of the water distributors and the water-regulating valves which control the water discharge to each compartment. Any section may be cut off for inspection or cleaning without interfering with the operation of any of the others. The water piping is designed for an additional installation of six towers of equal capacity.

Acceptance Test of High-Speed Poppet-Valve Engine—In the article on page 332 of the Sept. 7 issue of "Power," under this title, an unfortunate error occurred in Fig. 1, which should show an engine of the counterflow type with four valves, instead of the uniflow type. Engines of both the counterflow and uniflow design are built with poppet valves and this type of cylinder construction which allows for free expansion and contraction without temperature strains.

Boilers and Their Use in Chile

Following are some interesting facts from "Commerce Reports" as given by United States Consuls in Chile regarding boilers used in their districts.

Valparaiso—The boilers used in this district are usually of the Cornish, Galloway, Lancashire or Babcock & Wilcox type operating at a pressure 60 to 120 lb. Of the total of about 65,000 hp. developed in the country for industrial purposes, only 30,000 is by steam. The three boiler works in the country used material to the value of about $14,000 (U. S.) and produced $35,000 (U. S.) worth of boilers of the simplest type.

Iquique—Artificial heat of any kind is not required, and the few electric and nitrate works nearly all depend on internal-combustion engines for motive power. The Chile Exploration Co. is developing what is said to be the largest electric plant in South America. Oil-fired Babcock & Wilcox boilers will be used, and condensed sea water is the only available water supply. There are some 170 reduction works for nitrate of soda, each employing about five boilers of the Lancashire type, which are preferred on account of the high percentage of salt contained in the water. Steam is required for boiling tanks only.

The importation of boilers into Chile in 1911 and 1912 was valued at $60,000 and $111,000 respectively, of which the United States supplied only $333 and $5,500 worth. Coal is imported from England and Australia, the Chilean coal not being able to compete in quality or price. California fuel oil is now extensively used.

Recent Court Decisions

Digested by A. L. H. STREET

Judicial Interference with Rates—In dismissing a suit brought by an individual consumer of electricity to enjoin the St. Paul Gaslight Co. from continuing an established schedule of rates for lighting and power service, the Minnesota Supreme Court lately decided the following points:

Courts are empowered to determine whether rates established by a public-service corporation are reasonable, but are without authority to fix rates. If an adopted schedule is within the maximum prescribed by the proper legislative authority, an individual consumer has no standing in court to enjoin collection of the established rates on the ground of unreasonableness. The remedy of consumers for discrimination in rates by a public-service corporation is ordinarily an action at law for damages and not by injunction.

Protection of Water Rights—A city, in operating a water plant to supply its inhabitants with water, has no greater right to interfere with the flowage rights previously acquired by the owner of a downstream dam, without compensating him, than has a private owner of upstream lands, according to the decision of the Kansas Supreme Court in the case of Wallace vs. City of Winfield, 149 "Pacific Reporter," 693. It is also decided that where an owner of a private dam has maintained it for more than fifteen years it will be presumed that all upstream owners affected by it have consented to the maintenance of the dam in its established condition Speaking of defendant city's rights, as against plaintiff, who owns a downstream private dam, the supreme court says:

At the site of the proposed dam the defendant was a riparian owner by virtue of the purchase of land on each side of the river, and, while a riparian owner has the right to a reasonable use of the water for domestic and agricultural purposes, the city is not entitled to divert or take water from the river for the purpose of selling it to inhabitants of the city or to other persons remote from the stream The duty of supplying the inhabitants of the defendant city with water is imperative, and the necessity for obtaining water at this place may be pressing, but the city is vested with the power of eminent domain and may obtain the right to take water from the stream by condemnation and the payment of compensation to those who may be injured or from whom any property rights may be taken.

Warranty of Efficiency of Engines—When a written contract for the sale of an engine purports to contain all the conditions of the agreement bearing on the capacity of the engine, the seller will not be permitted to afterward rely upon verbal negotiations as extending the scope of this warranty. This is the substance of the decision announced by the Court of Appeals of Kentucky, in the case of Fairbanks, Morse & Co. vs. Manning & Combs, 175 "Southwestern Reporter," 1000. The plaintiffs sold a gasoline engine to the defendants, rating it at 8 hp. The only other terms concerning the efficiency of the engine expressed in the written contract of sale related to a warranty of workmanship and materials. An independent

clause in the agreement declared that there was no verbal understanding between the parties modifying the written contract. After delivery of the engine the defendants resisted payment of a balance due on the price on the ground that the engine failed to come up to verbal representations as to the amount of work it would do on a given hourly consumption of gasoline, but the Court of Appeals refused to permit the defendants to set up this oral warranty, stating that they should have seen that all warranties they intended to rely upon were inserted in the written contract.

PERSONALS

Perry Parker has withdrawn from the firm of Arthur D. Little, Inc., and established himself as a fuel engineer at 441 Milk St., Boston.

M. W. Ehrlich, formerly with Cole, Ives & Davidson, consulting engineers, has become editor of "Electrical Engineering," a monthly paper devoted to the field of electric practice.

Grant D. Bradshaw has resigned as chief engineer of the Cambria Steel Co., Johnstown, Penn., to take effect Oct. 1. He came to the Cambria company in 1910 as assistant steam engineer and was appointed steam engineer in the spring of 1911. Previously he was for one year with the Indiana Steel Co., Gary, Ind., and for five years with the Illinois Steel Co., Chicago. He is the inventor of the Bradshaw gas burner for blast furnace and producer gas, which is used at Cambria and has been adopted by a large number of other plants. He will become president of the Andrews-Bradshaw Co., sales engineers, of Pittsburgh. The company will make a specialty of steam and combustion equipment.

ENGINEERING AFFAIRS

The Detroit Engineering Society at its meeting on Sept. 8 was addressed by Dr. Charles Steinmetz, consulting engineer of the General Electric Co. His subject was "Electrical Engineering Developments."

The American Society of Refrigerating Engineers and the American Association of Refrigeration will hold a joint convention at San Francisco Sept. 21 to 25. Papers will be presented on "Oil-Engine Driven Raw Water Ice Manufacturing Plant," by Louis K. Doelling, New York; "Fruit Precooling and Precooling Plants," by W. C. Phillips, San Francisco; "The San Bernardino Precooling Plant," by C. M. Gay, Los Angeles; "Rate of Heat Transfer in Double Pipe Brine Coolers," by Van R. H. Greene and Fred Ophüls, New York. There will also be a general discussion on raw water ice making systems and on the power required to agitate brine in can ice tanks.

The Panama-Pacific Exposition has designated Sept. 25 as Machinery Day. A committee of 18 has been chosen from the exhibitors in the Machinery Palace, and with the aid of the exposition authorities this committee will arrange a special program of entertainment for that day. Since the date comes at the end of the week of the International Engineering Congress, the exhibitors in the Machinery Palace will make extraordinary effort to interest and entertain visiting engineers, students and the general public. The engineers are also invited to visit the Machinery Palace on the afternoon of Sept. 22, when the internal-combustion engine exhibitors will make special demonstrations of their apparatus. Sept. 22 is to be known as "Internal-Combustion Engine Day."

The Society to Promote the Science of Management has issued an invitation to all persons interested in scientific management and the work of Frederick W. Taylor to attend a memorial meeting to Mr. Taylor, to be held at 8:15 p.m., Friday, Oct. 22, in Houston Hall, University of Pennsylvania, Philadelphia. The program will include as speakers several who have been closely associated with Mr. Taylor in important periods of his career and important phases of his work. Distinguished leaders of the movement in foreign countries are preparing appreciations which will be read. Among these are Prof. A. Wallichs, Royal Polytechnical School, Aix-la-Chapelle; M. Henri Le Chatelier and M. Charles de Freminville, Paris; M. Charpy, director of the St. Jacques Steel Works; Prof. J. J. Sederholm, University of Helsingfors, Finland. The secretary of the society, Prof. Henry W. Shelton, Hanover, N. H., will supply further information.

NEW PUBLICATIONS

PURCHASING. By C. S. Rindsfoos, President of the United States Purchasing Corporation. Published by the McGraw-Hill Book Co., Inc., New York. Cloth, 6x9 in., 165 pages. Price, $2.

The salesman has been aided materially in his work by an extensive literature on the theory and practice of selling, but the man who must resist his sales attack has had to depend on native ingenuity, supplemented by experience gained in the daily work. Only recently has literature relating to the art of purchasing been available. The book under review is an example. The author has endeavored to formulate the principles governing the work of purchasing agents and to outline the general methods followed. We are first told how to obtain the right article, which is defined as that article best suited to meet the buyer's requirements. The right article is promptly delivered at the lowest price when the buyer is familiar with its intended use, with the article itself, the source of supply, and with the process of its manufacture. He must know something of the methods of testing and analyzing samples and should be familiar with market conditions that may be followed in the daily and trade papers. The author believes that the only sure method of getting the lowest price is by competitive bids, but he also discusses such buying methods as the lump sum, unit price and cost plus percentage, or fee. How to obtain the most favorable terms are summarized in these rules: "(1) If the buyer's credit is good, use this as a lever for a large cash discount or long dating. (2) If the seller's credit is poor, insist on a large cash discount. (3) Always arrange favorable terms, but make this the final concession which the seller must make to get the order." The careful buyer thus discusses the date of and the discount allowed for early payment only after the preliminary details have been settled. The terms are then used as an instrument of torture to squeeze the last drop of blood from the victim, in this case the salesman.

A chapter on personal characteristics and qualifications concludes by saying that the attributes of the "perfect purchasing agent" are the simple virtues desirable in other professions. Somewhat surprising, however, is the statement that broadmindedness and tactfulness are the qualities that the buyer most lacks and most needs. A section on strategy deals with methods of dickering, and while not recommending sharp practices, gives a number of them, so that the buyer may guard against their pitfalls. To apply strategy in buying involves a study of the salesman, the ferreting out of his weaknesses, the discerning of what is in his mind, and a knowledge of the policies of the house he represents. With this in hand, the purchasing agent can act or portray his own part accordingly. In a chapter on the legal aspects of purchasing are explained such elementary principles involved in purchasing goods, as the nature of a contract; requirements to be met before a contract can be legally enforced; method of making modifications; quantity, quality, and time clauses; fixing price terms; and the passing of title to the goods. Nearly half the text is devoted to the organization of the purchasing department. A report describing a proposed system for New York City is given, as are also a large number of forms to be used in executing and recording purchasing transactions. Altogether the book, written in a concise, readable style, should prove of value to anyone interested in buying supplies.

TRADE CATALOGS

The Wm. Powell Co., Cincinnati, Ohio. Booklet. Boiler valves. Illustrated, 22 pp., 3½x6½ in.

Betson Plastic Fire Brick Co., Rome, N. Y. Catalog. Plastic fire brick. Illustrated, 20 pp., 4x9 in.

A. D. Granger Co., 90 West St., New York. Bulletin No. 2. Oswego internally fired water tube boilers. Illustrated, 16 pp., 8x10½ in.

Busch-Sulzer Bros. Diesel Engine Co., St. Louis, Mo. Pamphlet. Appleton (Wis.) water works—combined filtration plant and Diesel engine pumping station. Illustrated, 20 pp., 7x10 in.

The Electric Storage Battery Co., Philadelphia, Penn. Hand Book H. L. Low voltage isolated electric light and power plants with the Chloride Accumulator. Illustrated, 52 pp., 6x9 in.

William Andrews, Inc., 120 Liberty St., New York. Catalog. The Deoleizer. Apparatus for removing cylinder oil from emulsified returns from a heating system or surface condenser. Illustrated, 7x10 in.